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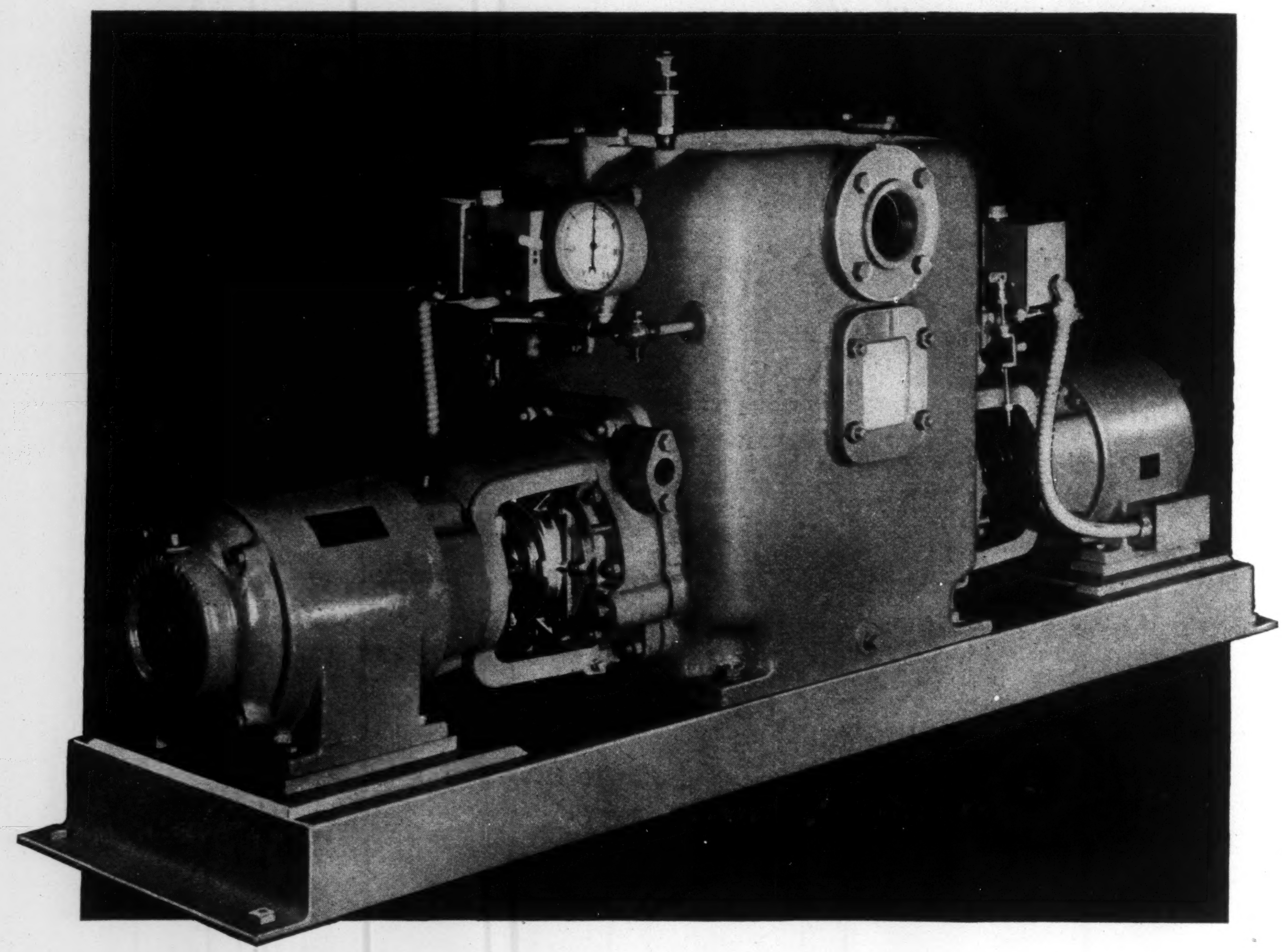

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Heating·Piping and Air Conditioning
Vol. 10, *No.* 11 ● *November,* 1938

Performance of Coils for Dehumidifying Air

By William Goodman*

When cooling and dehumidifying air by means of cold, wet surfaces, the condensation of moisture from the air complicates the problem of computing the performance of such surfaces. The equations and methods presented in this article have been developed in an effort to simplify these computations, and at the same time to illustrate the basic physical process involved. These methods and equations are not the same as those presented in the author's A.S.R.E. paper[1]; these new equations are analogous in form to the more familiar equations used for computing the performance of dry surfaces.[2] In addition, the final dry bulb temperature of the air can now be computed by means of a simple equation, whereas a graphical method was used in the A.S.R.E. paper.

The methods discussed here are equally applicable to direct expansion coils in which the temperature of the boiling refrigerant is substantially constant, and to counterflow coils in which the temperature of the water or brine changes. The computations for both types will be discussed in detail. The accuracy of these methods has been substantiated by tests on both direct expansion coils and counterflow water coils, the results of which will be discussed later.

The methods presented here permit the individual film coefficients of heat transfer to be determined from simple tests on actual coils without the use of thermocouple measurements of surface temperature. Although the method of determining these coefficients by test will be discussed later, no actual heat transfer coefficients will be presented because the coefficients for different coils vary. Such coefficients as are used in this article have been assumed only for the purpose of solving the various illustrative examples.

Heat Transfer Across a Wetted Surface

Equation 1 (derived in Appendix 2) is applicable to surfaces covered with moisture condensed from the air. It can be used to find the area of wetted surface needed to cool and dehumidify a given quantity of air, or to find the wet bulb temperature of the air leaving a coil of a given surface area. It can be used for direct expansion coils in which the temperature of the boiling refrigerant is constant, or for counterflow coils in which the temperature of a refrigerant such as water or brine rises as it flows through the coil.

$$H_W = A_W U_W d_M \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

In this equation, H_W is the total heat removed by the coil; it is the sum of the sensible and latent heat lost by the air. A_w is the *external* area of the coil. If the coil is finned this includes the area of the fins.

Although Equation 1 is similar in appearance to the familiar heat transfer formula for a dry surface, it differs from it in two important aspects:

1. U_W, the over-all coefficient for a wetted surface, is computed in a slightly different manner than the value of U_D for a dry surface.
2. d_M is not the mean temperature difference, but the mean enthalpy (total heat) difference.

These two quantities are discussed in detail in the sections that follow.

Equation 1 is derived in Appendix 2 by using an empirical linear relationship between h_s and t_s. For this reason this equation is not universal in its application. However, it does yield accurate results for the range of conditions ordinarily encountered.

Over-All Coefficient for a Wetted Surface

The value of U_D, the over-all coefficient for a dry surface, is computed from the air and refrigerant film coefficients by

$$\frac{1}{U_D} = \frac{1}{f_s} + \frac{B}{f_R} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

The value of U_W, the over-all coefficient for a wet surface, is computed by the following equation derived in Appendix 2.

New and simple formulas are developed by the author for computing the final dry and wet bulb temperatures of the air leaving a coil. Particularly noteworthy is the fact that the final wet bulb depression of the air is shown to depend upon the initial wet bulb depression of the air for both direct expansion and counterflow coils. This makes computation of the final dry bulb temperature the work of a moment

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*The Trane Co. Member of Board of Consulting and Contributing Editors.

[1] "Dehumidification of Air with Coils," *Refrigerating Engineering*, October, 1936, p. 225.

[2] "How to Solve Heat Transfer Problems," HEATING, PIPING AND AIR CONDITIONING, January, 1937, p. 11.

Part 1.

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Symbols

a = a numerical coefficient. See Table 1.
A_W = external area of wetted surface, sq ft.
B = ratio of external to internal surface area of coil.
c = humid specific heat of air (average $c = 0.243$).
d_L = large enthalpy (total heat) difference, Btu.
d_M = mean enthalpy (total heat) difference, Btu.
d_S = small enthalpy (total heat) difference, Btu.
D = wet bulb depression of the air, deg Fahr.
D_L = large temperature difference, deg Fahr.
D_M = mean temperature difference, deg Fahr.
D_S = small temperature difference, deg Fahr.
e = base of natural logarithms.
f_g = coefficient of heat transfer through air film, Btu per hr per sq ft per degree of temperature difference.
f_R = coefficient of heat transfer through refrigerant film, Btu per hr per sq ft per degree of temperature difference.
G = weight of air, lb per hr.
h = enthalpy (total heat) of humid air, Btu per lb of dry air.
h_R = enthalpy (total heat) of saturated air at a temperature equal to the refrigerant temperature, Btu per lb of dry air.
h_S = enthalpy (total heat) of saturated air at a temperature equal to the surface temperature, Btu per lb of dry air.
H_W = total heat lost by the air flowing past the wetted surface, Btu per hr.
M = a numerical factor. See Equations 7 and 11.
t = dry bulb temperature, deg Fahr.
t' = wet bulb temperature, deg Fahr.
t_R = refrigerant temperature, deg Fahr.
t_S = surface temperature of coil, deg Fahr.
U_D = over-all coefficient of heat transfer through dry surface of coil, Btu per hr per sq ft per degree of temperature difference.
U_W = over-all coefficient of heat transfer through wetted surface of coil, Btu per hr per sq ft per Btu of enthalpy difference.

Notes:

The subscript 1 refers to the initial condition of the air or refrigerant entering the coil. The subscript *2* refers to the final condition of the air or refrigerant leaving the coil.

All logarithms in formulas are to the base *e*.

$$\frac{1}{U_W} = \frac{c}{f_g} + B\frac{a}{f_R} \qquad \text{[3]}$$

The same values of the film coefficients f_g and f_R are used in both equations. The numerical value of these coefficients may be found either from the available literature on the subject or, better still, from tests on actual coils as described later. For a finned coil, the values of U_D and U_W represent the heat transferred per square foot of external surface.

In Equation 3, c is the specific heat of air. In an analogous manner, a might be called the specific heat of saturated air as it represents the average total heat removed in cooling one pound of saturated air one degree and condensing moisture. The values of a in [3] depend only upon the temperature of the refrigerant and are tabulated in Table 1. Although the use of an average value of a for any given refrigerant temperature is accurate only over a limited range, it is satisfactory in the usual range of conditions found in air conditioning work. This is discussed in more detail in Appendices 3 and 4. In the case of a refrigerant like water, the temperature of which changes as it flows through the coil, the value of a corresponding to the *initial* temperature of the refrigerant should be used. The reason for the variation in the value of a with the refrigerant temperature is explained in Appendices 3 and 4, where the method of computing these values is also explained.

Table 1—Values of *a* for Use in Equation 3

t_R	a	t_R	a
20	0.372	45	0.561
21	0.377	46	0.571
22	0.382	47	0.582
23	0.388	48	0.593
24	0.393	49	0.603
25	0.398	50	0.613
26	0.405	51	0.625
27	0.411	52	0.635
28	0.417	53	0.645
29	0.423	54	0.657
30	0.430	55	0.668
31	0.438	56	0.679
32	0.445	57	0.690
33	0.452	58	0.703
34	0.460	59	0.713
35	0.468	60	0.725
36	0.476	61	0.738
37	0.485	62	0.749
38	0.493	63	0.762
39	0.503	64	0.775
40	0.512	65	0.788
41	0.522	66	0.800
42	0.531	67	0.813
43	0.541	68	0.827
44	0.551	69	0.840

Factors c and a occur in [3] because the quantity of heat transferred through the outside air film of Fig. 1 is smaller than the quantity of heat transferred through the internal refrigerant film. This statement may sound absurd, but it is readily understandable in the light of the following explanation: Water vapor in the atmosphere exists in the form of superheated steam. No latent heat is released until this vapor has been cooled and condensed. However, the vapor cannot condense until it reaches the cold, wet surface. The water vapor can reach the cold, wet surface only by travelling through the outside air film. There is an actual flow of vapor from the main air stream to the cold surface by diffusion and convection. After the water vapor reaches the cold surface, it condenses and the latent heat released is transferred—together with the sensible heat surrendered by the air—through the metal wall of the tube to the refrigerant on the inside. Hence, it is evident that no latent heat is transferred through the outside air film; only sensible heat is transferred through this outside air film along with the actual water vapor that passes through the air film by diffusion and convection.

Example 1: For a given surface, $f_g = 12$, $f_R = 300$, and $B = 17$. The temperature of the boiling refrigerant inside the coil is 42 deg. Find: (a) The value of U_D for a dry surface. (b) The value of U_W for a wet surface.

Solution: (a) Using Equation 2,

$$\frac{1}{U_D} = \frac{1}{12} + \frac{17}{300} = 0.140$$

$U_D = 7.14$ Btu per deg temp. diff. per hr per sq ft of surface.

Fig. 1—Diagrammatic representation of surface films, and change in enthalpy (total heat) of humid air as it flows over a cold, wet surface

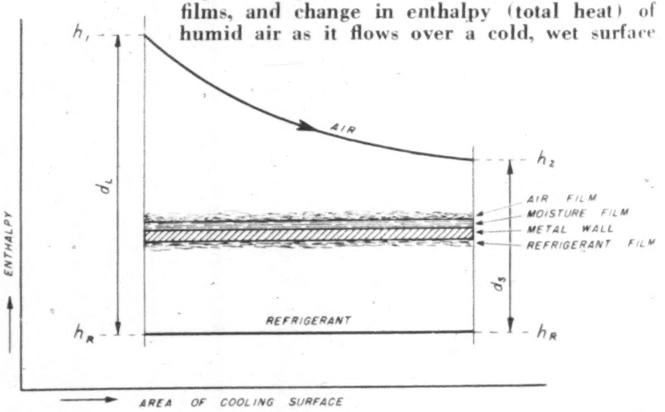

Table 2—Factors for Finding Log Mean Temperature Differences[a]

$\frac{D_S}{D_L}$	$\frac{D_M}{D_L}$	$\frac{D_M}{D_S}$	M	$\frac{D_S}{D_L}$	$\frac{D_M}{D_L}$	$\frac{D_M}{D_S}$	M
1	2	3	4	1	2	3	4
0.00	0.0000	∞	∞	0.50	0.7213	1.443	0.6932
0.01	0.2150	21.500	4.6052	0.51	0.7276	1.427	0.6735
0.02	0.2505	12.525	3.9120	0.52	0.7341	1.412	0.6539
0.03	0.2766	9.220	3.5065	0.53	0.7402	1.397	0.6350
0.04	0.2982	7.455	3.2189	0.54	0.7464	1.382	0.6163
0.05	0.3171	6.342	2.9957	0.55	0.7528	1.369	0.5977
0.06	0.3341	5.568	2.8136	0.56	0.7586	1.355	0.5800
0.07	0.3497	4.996	2.6596	0.57	0.7653	1.343	0.5619
0.08	0.3643	4.554	2.5257	0.58	0.7711	1.330	0.5447
0.09	0.3779	4.199	2.4079	0.59	0.7770	1.317	0.5277
0.10	0.3909	3.909	2.3026	0.60	0.7827	1.305	0.5110
0.11	0.4032	3.666	2.2072	0.61	0.7893	1.294	0.4941
0.12	0.4151	3.459	2.1199	0.62	0.7948	1.282	0.4781
0.13	0.4265	3.281	2.0399	0.63	0.8011	1.272	0.4619
0.14	0.4375	3.125	1.9657	0.64	0.8061	1.260	0.4466
0.15	0.4479	2.986	1.8976	0.65	0.8130	1.251	0.4305
0.16	0.4584	2.865	1.8326	0.66	0.8184	1.240	0.4154
0.17	0.4685	2.756	1.7716	0.67	0.8234	1.229	0.4008
0.18	0.4780	2.656	1.7156	0.68	0.8291	1.219	0.3859
0.19	0.4879	2.568	1.6601	0.69	0.8359	1.212	0.3709
0.20	0.4971	2.486	1.6094	0.70	0.8404	1.201	0.3570
0.21	0.5062	2.411	1.5607	0.71	0.8475	1.194	0.3422
0.22	0.5152	2.342	1.5140	0.72	0.8522	1.184	0.3286
0.23	0.5239	2.278	1.4697	0.73	0.8577	1.175	0.3148
0.24	0.5325	2.219	1.4272	0.74	0.8642	1.168	0.3009
0.25	0.5410	2.164	1.3863	0.75	0.8698	1.160	0.2874
0.26	0.5494	2.113	1.3470	0.76	0.8740	1.150	0.2746
0.27	0.5575	2.065	1.3094	0.77	0.8792	1.142	0.2616
0.28	0.5657	2.020	1.2729	0.78	0.8856	1.135	0.2484
0.29	0.5736	1.978	1.2378	0.79	0.8904	1.127	0.2359
0.30	0.5815	1.938	1.2039	0.80	0.8963	1.120	0.2231
0.31	0.5891	1.900	1.1712	0.81	0.9015	1.113	0.2108
0.32	0.5968	1.865	1.1394	0.82	0.9071	1.106	0.1984
0.33	0.6044	1.832	1.1086	0.83	0.9125	1.099	0.1863
0.34	0.6118	1.799	1.0788	0.84	0.9176	1.091	0.1744
0.35	0.6192	1.769	1.0498	0.85	0.9228	1.086	0.1626
0.36	0.6264	1.740	1.0217	0.86	0.9282	1.079	0.1508
0.37	0.6336	1.712	0.9944	0.87	0.9336	1.073	0.1392
0.38	0.6407	1.686	0.9677	0.88	0.9385	1.067	0.1279
0.39	0.6479	1.661	0.9416	0.89	0.9440	1.061	0.1165
0.40	0.6548	1.637	0.9163	0.90	0.9492	1.055	0.1054
0.41	0.6617	1.614	0.8916	0.91	0.9552	1.050	0.0942
0.42	0.6686	1.592	0.8675	0.92	0.9590	1.042	0.0834
0.43	0.6752	1.570	0.8442	0.93	0.9642	1.037	0.0726
0.44	0.6820	1.550	0.8211	0.94	0.9701	1.032	0.0619
0.45	0.6889	1.531	0.7984	0.95	0.9754	1.027	0.0513
0.46	0.6954	1.512	0.7766	0.96	0.9792	1.020	0.0409
0.47	0.7018	1.493	0.7552	0.97	0.9859	1.016	0.0304
0.48	0.7086	1.476	0.7338	0.98	0.9906	1.011	0.0202
0.49	0.7148	1.459	0.7134	0.99	0.9950	1.005	0.0101
				1.00	1.0000	1.000	0.0000

[a] This table is similar to Table 1 of "How to Solve Heat Transfer Problems," HEATING, PIPING AND AIR CONDITIONING, January, 1937.

(b) Referring to Table 1, for 42 deg, the value of a is 0.531. Using Equation 3,

$$\frac{1}{U_W} = \frac{0.243^{\dagger}}{12} + 17\,\frac{0.531}{300} = 0.05034$$

U_W = 19.9 Btu per Btu of enthalpy difference per hr per sq ft of surface.

The fact that the value of U_W is so much greater than the value of U_D is of no significance. The two values are not directly comparable because one is used with mean temperature difference, whereas the other is used with mean enthalpy (total heat) difference.

The resistance of the wall of the metal tube to the flow of heat has been neglected in Equations 2 and 3. The conductivity of the wall is so high compared to the conductivities of the surface films, and the wall of the tube is so thin, that its resistance to the flow of heat is negligibly small compared to the resistance of the two surface films.

Mean Enthalpy Difference for a Direct Expansion Coil

The value of d_M, the mean enthalpy difference, is found in exactly the same way as the mean temperature difference for a dry surface—namely, by means of the following commonly used equation (see Appendix 2):

$$d_M = \frac{d_L - d_S}{\log_e \dfrac{d_L}{d_S}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

The only difference is that the enthalpy (total heat) of the air is used instead of its dry bulb temperature.

The value of d_M can be computed quickly by means of Table 2, thus avoiding the use of Equation 4. In order to compute d_M, the values of d_L and d_S must first be found. Referring to Fig. 1, which has been drawn for a direct expansion coil in which the temperature of the boiling refrigerant is theoretically constant throughout the coil,

$$d_L = h_1 - h_R \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [5]$$

and

$$d_S = h_2 - h_R \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [6]$$

In these equations, h_1 is the initial enthalpy of the air entering the wet coil, h_2 is the enthalpy of the air leaving the wet coil, and h_R is the enthalpy of saturated air at a temperature equal to the temperature of the refrigerant.

After finding d_L and d_S, the value of d_M can be found by means of Table 2[††] as illustrated in Example 2 below.

Values of the enthalpy of saturated air for various temperatures can be found in the psychrometric tables.[3] For convenience in solving the examples, values of the enthalpy of saturated air through the range needed for this article are given in Table 4. The enthalpy of air is very nearly constant for any one wet bulb temperature.[3] Hence, throughout this article the enthalpy of air will be determined solely from its wet bulb temperature, and *vice versa.*

Example 2: Air at a wet bulb temperature of 72 deg is cooled to a final wet bulb temperature of 58 deg. The temperature of the refrigerant is 42 deg. Find d_M, the mean enthalpy difference.

Solution: From Table 4, for t'_1 = 72 deg, h_1 = 35.77 Btu; for t'_2 = 58 deg, h_2 = 25.09 Btu; for t_R = 42 deg, h_R = 16.15 Btu.

Using Equation 5, d_L = 35.77 — 16.15 = 19.62 Btu.

Using Equation 6, d_S = 25.09 — 16.15 = 8.94 Btu.

$$\frac{d_S}{d_L} = \frac{8.94}{19.62} = 0.456$$

Referring to Table 2 for $\frac{d_S}{d_L} = 0.456$, $\frac{d_M}{d_L} = 0.693$.

d_M = 0.693 × 19.62 = 13.60 Btu, mean enthalpy difference.

If the value of d_S/d_L is *greater* than 0.6, the arith-

[†] Strictly speaking, the value of c, the humid specific heat of air, increases with the dewpoint temperature. However, the specific heat of humid air changes very slowly with increasing dewpoint temperatures. Notice in the psychrometric tables[3] that for dewpoint temperatures below 60 deg, the specific heat of humid air is less than 0.246. Inasmuch as the dewpoint temperatures commonly encountered are less than about 60 deg, the use of an average value of 0.243 is satisfactory for most purposes.

[††] The values in Table 2 can be used to figure either mean temperature difference or mean enthalpy difference. Disregard the fact that capital letters are used at the head of each column in Table 2. This table is similar to Table 1 of the article mentioned in Footnote 2.

[3] "New Tables of the Psychrometric Properties of Air-Vapor Mixtures," HEATING, PIPING AND AIR CONDITIONING, January, 1938, p. 1.

metical average of d_S and d_L will deviate by less than 2 per cent from the mean temperature difference computed by means of Table 2.

If the pressure were the same throughout a direct expansion coil, the temperature of the boiling refrigerant would be constant throughout the coil. However, because of the small pressure drop of the liquid while flowing through the tubes, the temperature of the boiling refrigerant usually varies slightly. For computing the mean enthalpy difference of a direct expansion coil, use the saturation temperature of the refrigerant corresponding to the average pressure *inside the tubes* of the coil. When finding this average pressure, the pressure drop in the liquid header or other liquid distributing device should be ignored; only the change in pressure while the liquid is actually inside the tubes forming the coil should be considered.

Any superheating of the refrigerant that takes place inside the coil should be ignored when computing the mean enthalpy difference. Even if some superheating does occur, only the average temperature inside the tubes of the coil should be used. In a counterflow coil (to be discussed later) the computation of the mean enthalpy difference is based on the temperatures of the water (or other refrigerant) entering and leaving the coil—not on the average temperature of the refrigerant. The reason for the difference in computation procedure for direct expansion coils and counterflow coils is this: As the water in a counterflow coil rises in temperature, it absorbs an equal amount of heat for each degree of temperature rise. Thus if water rises 10 deg in temperature while flowing through a counterflow coil, it absorbs 10 per cent of the total heat removed by the coil for each degree rise in temperature. On the other hand, if a refrigerant vaporizes inside a coil at substantially constant temperature and is then superheated 10 deg, only a very small percentage of the total heat removed by the coil will be absorbed by the superheated vapor. By far the major part of the heat transfer takes place while the refrigerant is boiling at constant temperature; the amount absorbed by the superheating of the vapor is negligibly small by comparison. Hence, inasmuch as the refrigerant absorbs practically all of the heat while its temperature is constant, only this constant temperature should be used in computing either the mean temperature difference or the mean enthalpy difference. Furthermore, the derivation of Equations 1 and 4 is based on the heat transfer process just described, namely that:

1. In the case of a vaporizing refrigerant, the temperature of the fluid remains constant during the heat transfer process.
2. In the case of a refrigerant whose temperature changes (such as water or brine), equal amounts of heat are absorbed for each degree change in temperature.

Area of Wetted Surface Required for a Direct Expansion Coil

After finding the values of U_W and d_M as described in the preceding sections, Equation 1 can be used to find the area of wetted surface needed to cool and dehumidify a given quantity of air.

Example 3: Air at an initial wet bulb temperature of 72 deg is to be cooled to a final wet bulb temperature of 58 deg with a refrigerant at 42 deg. The total quantity of air to be cooled is 12,500 cfm. For the particular coil in question, f_s = 12, f_R = 300, and B = 17. Find the total area of wetted surface required to cool and dehumidify the air.

Solution: From Example 1, U_w = 19.9. From Example d_M = 13.60 Btu.

G = 12,500 × 4.5 = 56,250
(60 × 0.075 = 4.5, where 0.075 is th weight of 1 cu ft of standard air.)

$H_W = G(h_1 - h_2)$

= 56,250 (35.77 — 25.09) = 600,800 Btu pe hr.

Table 3—Factors for Finding Log Mean Temperature Differences

M	D_M/D_L	D_M/D_S	D_S/D_L	M	D_M/D_L	D_M/D_S	D_S/D_L
1	2	3	4	1	2	3	4
0.00	1.0000	1.000	1.0000	0.75	0.7034	1.489	0.4724
0.01	0.9950	1.010	0.9901	0.76	0.7006	1.498	0.4677
0.02	0.9901	1.010	0.9802	0.77	0.6974	1.506	0.4630
0.03	0.9850	1.017	0.9704	0.78	0.6943	1.515	0.4584
0.04	0.9803	1.020	0.9608	0.79	0.6913	1.523	0.4539
0.05	0.9754	1.026	0.9512	0.80	0.6883	1.532	0.4493
0.06	0.9708	1.030	0.9418	0.81	0.6854	1.541	0.4448
0.07	0.9661	1.036	0.9324	0.82	0.6827	1.550	0.4404
0.08	0.9611	1.041	0.9231	0.83	0.6794	1.558	0.4361
0.09	0.9562	1.047	0.9139	0.84	0.6763	1.567	0.4317
0.10	0.9516	1.052	0.9048	0.85	0.6737	1.577	0.4274
0.11	0.9470	1.057	0.8958	0.86	0.6709	1.585	0.4232
0.12	0.9423	1.062	0.8869	0.87	0.6679	1.594	0.4189
0.13	0.9377	1.068	0.8781	0.88	0.6648	1.603	0.4148
0.14	0.9332	1.074	0.8694	0.89	0.6621	1.612	0.4107
0.15	0.9287	1.079	0.8607	0.90	0.6593	1.622	0.4066
0.16	0.9241	1.084	0.8521	0.91	0.6565	1.631	0.4026
0.17	0.9196	1.090	0.8437	0.92	0.6538	1.641	0.3985
0.18	0.9152	1.096	0.8353	0.93	0.6511	1.651	0.3945
0.19	0.9107	1.101	0.8270	0.94	0.6484	1.660	0.3906
0.20	0.9063	1.107	0.8187	0.95	0.6456	1.669	0.3867
0.21	0.9020	1.113	0.8106	0.96	0.6430	1.679	0.3829
0.22	0.8976	1.119	0.8025	0.97	0.6401	1.689	0.3791
0.23	0.8934	1.124	0.7945	0.98	0.6372	1.698	0.3753
0.24	0.8891	1.130	0.7866	0.99	0.6347	1.708	0.3716
0.25	0.8848	1.136	0.7788	1.00	0.6321	1.718	0.3679
0.26	0.8805	1.142	0.7711	1.02	0.6269	1.738	0.3606
0.27	0.8764	1.148	0.7634	1.04	0.6217	1.759	0.3535
0.28	0.8721	1.154	0.7558	1.06	0.6166	1.780	0.3465
0.29	0.8680	1.160	0.7483	1.08	0.6115	1.801	0.3396
0.30	0.8639	1.166	0.7408	1.10	0.6065	1.822	0.3329
0.31	0.8599	1.172	0.7334	1.12	0.6016	1.844	0.3263
0.32	0.8558	1.178	0.7261	1.14	0.5967	1.866	0.3198
0.33	0.8517	1.185	0.7189	1.16	0.5918	1.888	0.3135
0.34	0.8477	1.191	0.7118	1.18	0.5871	1.911	0.3073
0.35	0.8437	1.197	0.7047	1.20	0.5823	1.933	0.3012
0.36	0.8399	1.204	0.6977	1.22	0.5777	1.957	0.2952
0.37	0.8359	1.210	0.6907	1.24	0.5731	1.980	0.2894
0.38	0.8321	1.217	0.6839	1.26	0.5686	2.004	0.2837
0.39	0.8280	1.223	0.6771	1.28	0.5641	2.029	0.2780
0.40	0.8242	1.230	0.6703	1.30	0.5596	2.053	0.2725
0.41	0.8204	1.236	0.6637	1.32	0.5552	2.078	0.2671
0.42	0.8166	1.243	0.6570	1.34	0.5509	2.104	0.2618
0.43	0.8129	1.250	0.6505	1.36	0.5466	2.130	0.2567
0.44	0.8091	1.256	0.6440	1.38	0.5423	2.156	0.2516
0.45	0.8052	1.263	0.6376	1.40	0.5382	2.182	0.2466
0.46	0.8016	1.270	0.6313	1.42	0.5340	2.209	0.2417
0.47	0.7980	1.277	0.6250	1.44	0.5299	2.236	0.2369
0.48	0.7941	1.283	0.6188	1.46	0.5258	2.264	0.2322
0.49	0.7906	1.291	0.6126	1.48	0.5219	2.293	0.2276
0.50	0.7869	1.297	0.6065	1.50	0.5179	2.321	0.2231
0.51	0.7831	1.304	0.6006	1.55	0.5083	2.395	0.2122
0.52	0.7797	1.311	0.5945	1.60	0.4988	2.471	0.2019
0.53	0.7762	1.319	0.5886	1.65	0.4897	2.550	0.1921
0.54	0.7728	1.326	0.5827	1.70	0.4807	2.632	0.1827
0.55	0.7691	1.333	0.5770	1.75	0.4721	2.717	0.1738
0.55	0.7658	1.341	0.5712	1.80	0.4638	2.806	0.1653
0.57	0.7621	1.347	0.5656	1.85	0.4555	2.897	0.1572
0.58	0.7587	1.355	0.5599	1.90	0.4476	2.992	0.1496
0.59	0.7554	1.363	0.5543	1.95	0.4398	3.092	0.1423
0.60	0.7521	1.370	0.5488	2.00	0.4323	3.195	0.1353
0.61	0.7483	1.377	0.5435	2.10	0.4179	3.413	0.1225
0.62	0.7453	1.385	0.5379	2.20	0.4041	3.647	0.1108
0.63	0.7421	1.394	0.5325	2.30	0.3912	3.902	0.1003
0.64	0.7388	1.401	0.5273	2.40	0.3789	4.177	0.0907
0.65	0.7356	1.409	0.5219	2.50	0.3672	4.473	0.0821
0.66	0.7320	1.416	0.5169	2.60	0.3560	4.794	0.0743
0.67	0.7286	1.424	0.5118	2.70	0.3455	5.141	0.0672
0.68	0.7258	1.433	0.5066	2.80	0.3354	5.515	0.0608
0.69	0.7225	1.441	0.5015	2.90	0.3258	5.922	0.0550
0.70	0.7194	1.449	0.4966	3.00	0.3167	6.361	0.0498
0.71	0.7161	1.456	0.4916	3.20	0.2998	7.354	0.0408
0.72	0.7129	1.465	0.4868	3.40	0.2843	8.518	0.0334
0.73	0.7097	1.473	0.4819	3.60	0.2702	9.889	0.0273
0.74	0.7064	1.481	0.4771	3.80	0.2573	11.502	0.0224

$$A_W = \frac{H_W}{U_W d_M} \qquad \text{[1]}$$

$$= \frac{600{,}800}{19.9 \times 13.60} = 2220 \text{ sq ft of wetted external surface.}$$

Final Wet Bulb Temperature of Air Leaving Wetted Surface of a Direct Expansion Coil

Equation 1 can also be used to find the final wet bulb temperature of the air leaving a coil of a given area. For this purpose, the value of the factor M defined by the following equation must first be found:

For finding the final *wet* bulb temperature of air leaving *direct expansion* coils,

$$M = \frac{A_W U_W}{G} \qquad \text{[7]}$$

After finding the value of M for a given coil, the value of d_S/d_L can be found by referring to either Table 2 or 3*. From this the final enthalpy and, therefore, the final wet bulb temperature of the air can be determined.

Example 4: Take the conditions of Example 3 except that the final wet bulb temperature of the air is unknown and is to be found for a coil of 1600 sq ft of wetted external area. Find the final wet bulb temperature of the air leaving the coil and the total heat that can be removed from the air by the coil.

Solution: Using Equation 7, $M = \frac{1600 \times 19.9}{56{,}250} = 0.566.$

Referring to Table 3, for $M = 0.566$, $d_M/d_L = 0.7636$.
From Example 2, $d_L = 19.62$ Btu.
Therefore, $d_M = 0.7636 \times 19.62 = 14.98$ Btu.
From Example 1, $U_W = 19.9$ Btu.

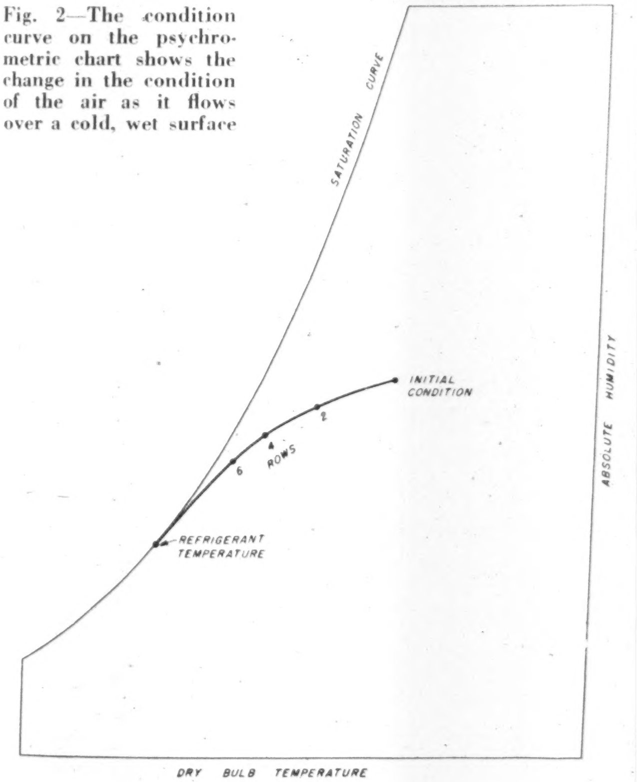

Fig. 2—The condition curve on the psychrometric chart shows the change in the condition of the air as it flows over a cold, wet surface

The total heat removed by the coil is found by means of Equation 1,

$$H_W = 1600 \times 19.9 \times 14.98 \qquad \text{[1]}$$
$$= 477{,}000 \text{ Btu per hr.}$$

From Example 3, $G = 56{,}250$ lb per hr of air. Heat removed per lb of air flowing through the coil is

$$\frac{H_W}{G} = \frac{477{,}000}{56{,}250} = 8.48 \text{ Btu per lb of air.}$$

From Example 2, $h_1 = 35.77$ Btu per hr.
Therefore, $h_2 = 35.77 - 8.48 = 27.29$ Btu.
Referring to Table 4, for $h_2 = 27.29$ Btu, $t'_2 = 61.3$ deg, final wet bulb temperature of the air leaving the coil.

Final Dry Bulb Temperature of Air Leaving a Direct Expansion Coil

For any given area of wetted surface and any given air velocity,** the *final wet bulb depression of the air is always a definite fraction of the initial wet bulb depression.* This is evident from Equation 8, which is derived in Appendix 5.

$$\frac{D_2}{D_1} = e^{-M} \qquad \text{[8]}$$

$$D_1 = t_1 - t'_1 \qquad \text{[9]}$$

$$D_2 = t_2 - t'_2 \qquad \text{[10]}$$

$$M = \frac{A_W f_g}{Gc} \qquad \text{[11]}$$

Also from [8] and [11],

$$\text{Log } \frac{D_1}{D_2} = \frac{A_W f_g}{Gc} \qquad \text{[8a]}$$

As shown in Example 5, the value of D_2 can be computed by means of Table 3, without the necessity of using Equation 8. The value of D_2/D_1 can be found in Column 4 (D_S/D_L) of Table 3 after first computing the value of M by means of Equation 11. From this D_2 can be computed. Then Equation 10 can be used to find t_2, the final dry bulb temperature of the air leaving the coil.

Example 5: Take the conditions of Example 4. The initial dry bulb temperature of the air entering the coil is 90 deg and the initial wet bulb is 72 deg. Find the final dry bulb temperature of the air leaving the coil.

Solution: From Example 4, $t'_2 = 61.3$ deg. By Equation 9, $D_1 = 90 - 72 = 18$ deg.

Using Equation 11, $M = \frac{1600 \times 12}{56{,}250 \times 0.243} = 1.40$

Referring to Table 3 for $M = 1.40$, $\frac{D_2}{D_1} = 0.2466$.

$$D_2 = 0.2466 \times 18 = 4.4 \text{ deg.}$$

Therefore, using Equation 10,

$$t_2 = 61.3 + 4.4 = 65.7 \text{ deg.}$$

If all of the conditions of Example 4 had been kept constant except that the area of the coil was varied by increasing the rows of tubes composing the coil, the final dry and wet bulb temperatures of the air leaving these various rows could be found by the methods illustrated in Examples 4 and 5. Then if the final condition of the air leaving these coils of different numbers of rows of tubes were plotted on a psychrometric chart, all of these points would fall on a curve of the form illustrated in Fig. 2. This curve is called the "condition curve" because it shows the condition of the air at any point in its passage over a cold wetted surface. As the number of rows of tubes is increased, the point repre-

[Continued on page 707]

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*When the value of M is known, Table 3 is easier to use. Tables 2 and 3 are identical, except for the fact that D_S/D_L is the independent variable in Table 2, whereas in Table 3, M is the independent one.

**Because both f_g and G depend upon the air velocity.

SERVICING AIR CONDITIONING JOBS

High Pressure in Condensers

By Melvin A. Ramsey*

The author has given thought to a better way of eliminating air from refrigeration condensers without wasting more refrigerant than absolutely necessary. He has used the method described here with satisfactory results and offers it as of interest to other readers

MOST instructions for operation of "Freon" refrigeration systems note that a condenser pressure above the normal expected, or above the amount which has been usual, may be caused by air in the system. To find out whether or not this is the cause, the condenser is purged a little to see if an improvement results, or, possibly, an attempt is made to see whether or not the gas discharged from the condenser is "Freon." Since even if air is present "Freon" will be mixed with it, this is somewhat difficult. In any case there is a possibility of wasting "Freon" when air is not the reason for the high pressure.

Before doing anything about unusually high condenser pressure, the cause should be determined. This can be done without opening the condenser in most cases.

With the compressor stopped, let the condenser water continue to flow. This may mean opening very wide or bypassing the automatic water valve, if one is used. An accurate gage should indicate the condenser pressure. It is well to shut the inlet and outlet refrigerant valves. After the water has flowed for some little time, its temperature and the pressure in the condenser will both be about steady. Determine the saturation pressure of the refrigerant corresponding to the condensing water temperature (the leaving and entering water temperature at this time will, of course, be equal).

If the actual pressure in the condenser is much above this saturation pressure, some other gas beside "Freon" must be present. In case this is true, the condenser must be purged. A valve should be opened at the very top of the condenser, and the gas discharged slowly (in comparison with the total volume) until the pressure is about 2 or 3 lb above the pressure which should exist or, if a level gage is provided, until the level moves, indicating the start of boiling. The rate of discharge for a large condenser can be more rapid than that for a smaller one. The rate is of importance, because the "Freon" probably tends to separate and lie below the air, and too rapid venting may result in a loss of more "Freon."

There is a question of how much stratification takes place because of the very dense "Freon." It would be interesting to see the results of any tests which may have been made of the percentage of "Freon" in the mixture which is eliminated in purging.

A small condenser to which the mixture could be pumped from the larger one instead of purging directly to atmosphere would make it possible to increase the total pressure and also that on the "Freon," thus making it possible to condense more "Freon" out of the mixture before purging. If the small condenser were arranged to have chilled water instead of well water circulated through it, more "Freon" could be removed from the mixture before purging to atmosphere. The liquid could then be allowed to go back to the main condenser by having the bottoms and tops of both condensers connected and raising the smaller one above the larger so that the "Freon" would return by gravity.

The pressure in the small condenser can be increased by using a small standard "Freon" condensing unit (possibly ¼ hp) for a very large system and a small circulator with connections to the chilled water system. Since air need not be eliminated often such a unit would probably not be permanently installed but could be a part of the equipment of the service engineer.

(On direct expansion jobs where no chilled water is available, the liquid refrigerant supplied by the small compressor could be expanded directly into the coil of the small condenser. In this way, the "Freon" and air mixture would be chilled to a very low point in the secondary condenser, thus condensing out the "Freon" from the mixture. This method can also be used on chilled water systems because it eliminates the need of circulating chilled water through the small secondary condenser.)

Allow the condenser water to flow for several minutes until pressure and temperature are each stable. At the end of this time, if the pressure in the condenser is not appreciably above the saturation pressure corresponding to the water temperature, it is evident that air is not the trouble. A check should then be made for adequate water circulation. Often this is difficult to measure, but it can usually be approximated from inlet and outlet condenser water temperatures and data on the compressor capacity under the conditions prevailing.

If no air is present and sufficient water is circulating, the condenser is too small or its capacity is limited by dirty or clogged tubes. If the system has been in satisfactory operation and is just beginning to give trouble, it must of course be adequate—assuming the load on it has not been increased in some way. If after the above checks the capacity is also known to be sufficient, the condenser should be examined as to need of cleaning.

The use of the method outlined avoids waste of "Freon" when purging is unnecessary. If purging is necessary, it is possible to know about when it is completed. The method has been used with very satisfactory results, and seems unusually simple to follow.

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Inspection of Welded Piping

By William D. Halsey*

With the increased use of welding in the construction of piping has come the demand for inspection during construction, for fusion welded joints are dependent to a far greater degree than are mechanical joints upon the method of making them and upon the individuals who do the work. . . . The author discusses the function of the inspecting agency, the importance of setting forth a written procedure for welding, and the guided bend test given in the standard qualification procedure of the American Welding Society

UNTIL recent years customary piping construction—at least for the higher pressure piping—has been by threading, van stoning or some similar mechanical fastening; seldom in the past was an inspection made by an independent agency of pipe lines constructed by such means. However, within the past few years the welding of pipe lines has grown very rapidly and with it there has come a demand on the part of purchasers for inspections during construction.

Without doubt, pipe lines constructed with the various types of mechanical joints were in many cases not perfect, but because of the nature of the joint some degree of imperfection was permissible. If defects were present, the most probable result would be leakage at a joint, which would make repairs mandatory before a violent failure took place. Fusion welded joints in pipe lines, however, are dependent to a far greater degree than are mechanical joints upon the method of making them, and upon the individuals who do the work. Whereas in a mechanical pipe joint, leakage gives warning of an impending failure, in a fusion welded joint a complete failure may occur almost immediately following the development of a leak.

Mechanical joints for the fabrication of pipe lines are open to some objections when used for high pressure and high temperature service which do not apply to fusion welded construction provided the job has been properly designed for fusion welding and the welding itself has been properly done. It is because fusion welding has been applied for severe service in pipe lines where a failure might mean the loss of many thousands of dollars, and because the adequacy of a welded joint to withstand the severe service depends to a very high degree on the procedure of welding used and the competency of the men who do the work, that inspections began in the field of high pressure power piping. The idea has been extended to the lower pressure field, and in some cases even to the heating field. Most inspections have been confined to the quality of the welding, but other items are also covered.

Important Items in Inspections

The Code for Pressure Piping is being referred to more and more in writing specifications for pressure piping. This code has sections relating to power piping, to gas and air piping, to oil piping, and to piping for district heating. The code, however, does not at this time have any specific requirements for inspections during construction such as are found in the A.S.M.E. power boiler and unfired pressure vessel codes, and what should be done in the way of inspecting under the piping code is, therefore, largely a matter of choice. However, it has seemed that there are some items which are of importance, such as the quality of the material used and its adaptability to the service intended, and the proportions of the flanges and the bolts where used. The adequacy of the welding, both from the standpoint of the method used and the competency of the welding operators, is undoubtedly a most important point to be covered. The piping code calls for a hydrostatic test of the completed line or its component parts and it is considered essential that an inspection be made at that time.

It must be borne in mind that when a purchaser specifies inspections by an independent agency of a fusion welded pipe line during construction, the inspecting agency is acting as an agent for the purchaser. While such inspections have been called for on fusion welded pipe lines primarily because of the welded construction, at the same time it is considered that good workmanship will be expected and that the letter and the spirit of the piping code or the specifications will be adhered to. If any change is to be made it should be only with the consent of the purchaser.

It is agreed by those who have had experience in the application of welding for severe service that a rather carefully developed technique or procedure for welding must be used. While the employment of a competent welding operator is necessary and while there are many intelligent operators who can and do give great assistance in developing proper procedures for welding, it cannot be considered that mere years of experience on the part of a welding operator is a safe criterion to use for his ability or for his competency to advise as to proper methods of welding. As a matter of fact, welding tests frequently illustrate that the welding operator who has had the most years of experience may be the one who does the poorest work. This is not always a reflection on the welding operator, because there have been instances where workmen have merely been furnished equipment with no instruction as to how to use it. Such workmen have frequently fallen into improper ways of doing welding and have never realized the errors of those ways nor had them called to their attention.

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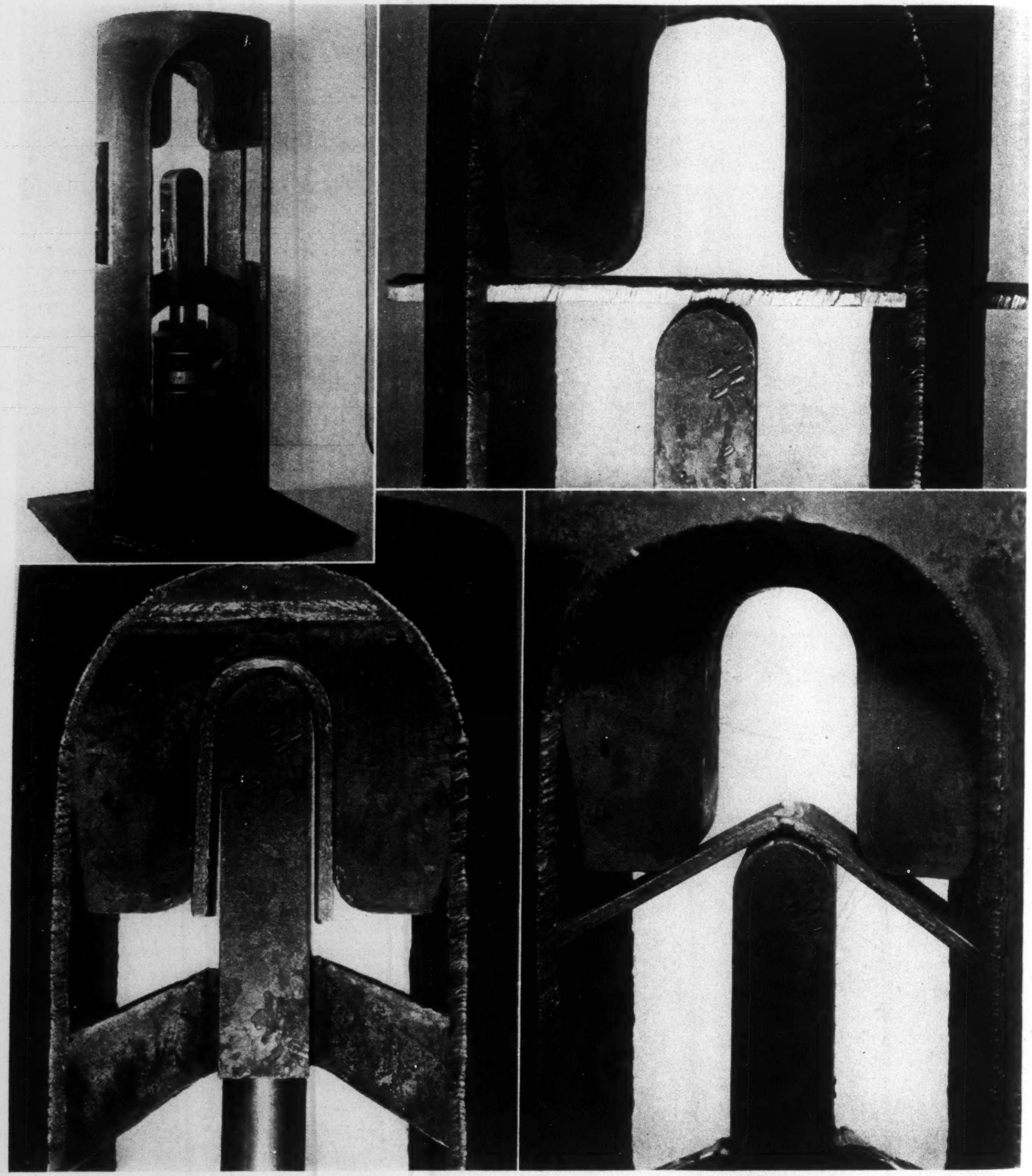
Development of Welding Procedure

The development of a proper procedure for welding requires that consideration be given to the preparation of the ends of the pipe, and the spacing to be provided. An investigation of the most satisfactory welding rod or electrode should be made. Such an investigation does not necessarily mean exhaustive testing but does mean consulting with welding engineers who are in a position to render an opinion. The size of the electrode and the "heat," or the amperage during welding, must be given consideration. In oxy-acetylene welding, careful thought must be given to whether the welding should be done by the backhand or forehand technique. In many cases,

***Upper left*—The guided bend tester is a rather simple device using a hydraulic jack. . . . *Upper right*—Close-up view of specimen in guided bend tester at start of test. . . . *Lower left*—Completion of test of an acceptable specimen. . . . *Lower right*—A specimen which failed to pass the test**



peening of each successive bead of welding is considered advisable if not absolutely necessary. Many feel that preheating of even ordinary low carbon steel pipe is a highly advisable thing to do and in fact such preheating becomes mandatory for some kinds of material. Consideration must also be given to the necessity or desirability of stress relieving and how it should be accomplished.

All such essential points, some of which are enumerated above, may be set forth in a written procedure for welding and the piping contractor or fabricator should train his welding operators to follow that method of welding in all its essential details. It must be borne in mind that the contractor or fabricator, and not the individual welding operator, is responsible for the completed work and the purchaser will look to the fabricator or contractor for correction of any trouble that develops rather than to the welding operator who may have been employed. For this reason it is essential that the fabricator or contractor understand fully why all the items in the welding procedure must be adhered to, and he must see that he has the work of welding under control and that it is properly supervised.

Qualifying the Welding Operator

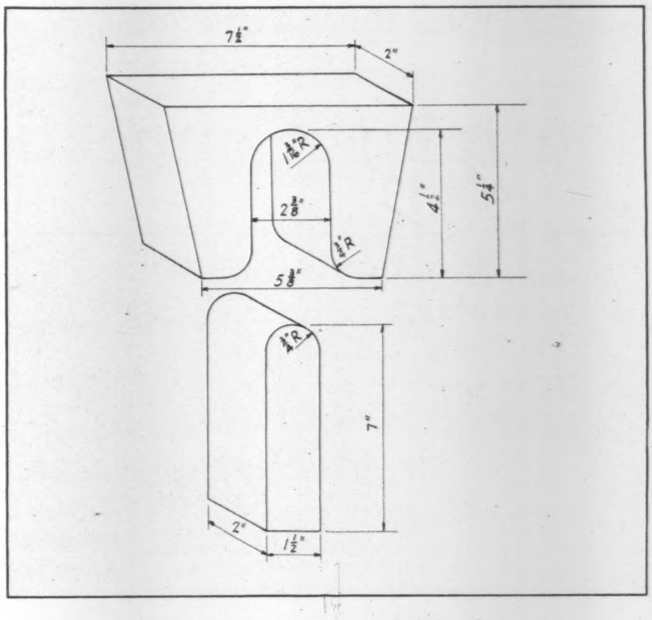
As stated above, competent welding operators are necessary. By the term "qualified welder" is meant an operator who has demonstrated his ability to make a weld meeting certain requirements under a certain given procedure and conditions. At one time, and rather unfortunately for the welding industry, there was no recognized method of testing welding operators. A number of years ago the thought was expressed that the ability of every welding operator should be examined and the earlier recognized testing procedure called for tests for tensile strength, ductility, and soundness.

A later idea expressed the thought that the tensile strength and ductility of weld metal depended upon the materials placed in the hands of the welding operator and that his competency could be judged merely by tests for soundness of the welds or his ability to fuse the weld metal to the side walls of the welding groove. This thought resulted in a series of tests for soundness which involved breaking open the specimen welded joint. The difficulty presented by this method was that of evaluating the broken specimen. What one examiner might feel was a satisfactory degree of soundness, another examiner might feel was unsatisfactory. In other words there was no accepted yardstick which could be applied to this method of examining a welding operator's ability.

The Guided Bend Test

Early this year the American Welding Society published its standard qualification procedure giving a new method of testing a welding operator. This method, using the "guided bend test," is rather simple to apply and involves no question of judgment on the part of the examiner inasmuch as it is a "go-or-not-go" test.

In using this method of testing, the welding operator makes a weld between two pieces of pipe with a wall thickness of ⅜ in. and with a diameter of not less than 6 in. Larger diameter pipe may be used, the essential requirement being that the wall thickness be ⅜ in. One or two specimens or test welds are made, either in the vertical or horizontal fixed position or both. The pieces of pipe for this test may be relatively short, say about 5 in. each, and should be prepared on one end for welding according to the specification that has been developed for the welding procedure to be followed. When a welding operator has completed the welding of these joints, four specimens or strips 1½ in. wide are cut from each. In the case of the joint welded in the horizontal fixed position, specimens are taken from the top, bottom, and the two sides. In the specimen welded in the vertical fixed position, the four specimens are taken at 90 deg from each other.



U-block and plunger for guided bend tester

The outside reinforcement of the weld on these specimens is removed, by machining or grinding, flush with the surface of the pipe. The specimens are then placed in a "guided bend tester," a rather simple device using a hydraulic jack which may be constructed at small cost, such as the one illustrated here. This tester has in it a U-block and plunger machined to the contour as shown. For specimens ⅜ in. thick, a five ton hydraulic jack is adequate. The specimen is forced into the U-block until it takes the contour of that form. Some of the specimens are tested with the root and some with the face of the weld in tension. To be acceptable, the specimen must withstand this bending without developing a crack more than ⅛ in. long.

In general, such a test in ⅜ in. material qualifies the welding operator up to and including a maximum thickness of ¾ in. For material in excess of ¾ in. thickness, the operator makes an additional test weld or welds in the maximum thickness for which he is to be qualified except that the maximum thickness of the test weld need not exceed 1½ in. The specimens cut from these heavier wall test welds are made ⅜ in. wide rather than 1½ in. and bent sidewise rather than by face and root.

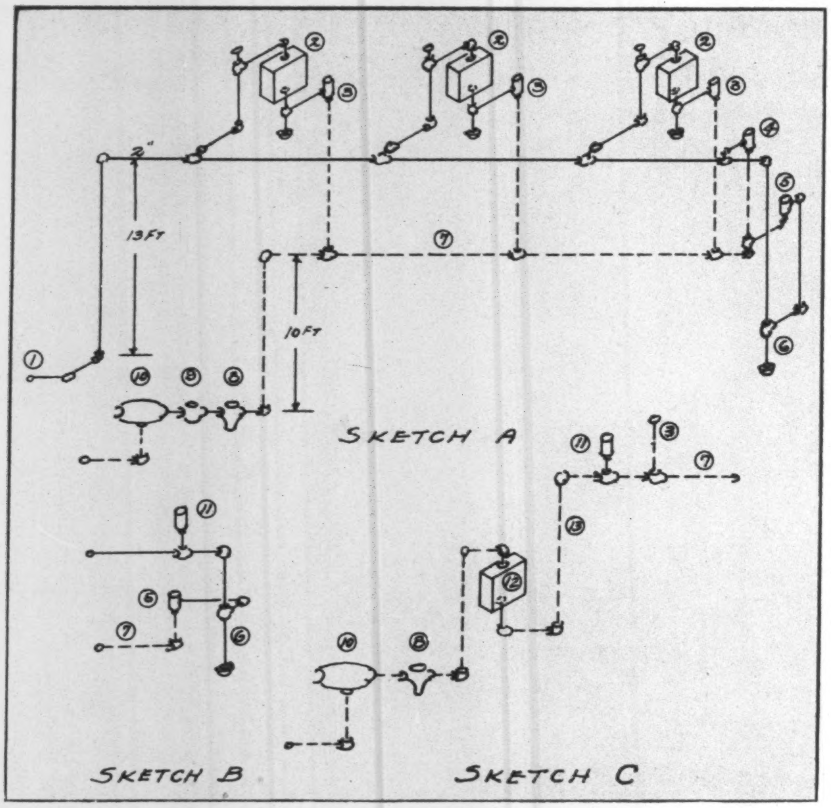
It will be recognized that this method of testing is one for which testing equipment can be readily provided and such equipment easily can be transported. The method of testing is conclusive and avoids questions of

[Concluded on page 707]

Reader Asks for Advice on

Piping Connections for Unit Heaters

Answered by Samuel R. Lewis*



A READER of H. P. & A. C. submits a sketch of a heating system supplied with steam from a central public service plant. He states that steam enters at a very uniform pressure of 1 lb and that the condensate is wasted to the sewer.

Sketch *A* indicates in a general way the manner in which he proposed to install the system, using three unit heaters, each having an electric fan. He submits an alternate sketch *B*, and asks which arrangement at the end of the supply main will give the better results.

Analyzing sketch *A*, steam enters at *1* and the 2 in. supply main rises 13 ft to an elevation near the ceiling, then grades downward to the right, terminating at *5* after serving three small unit heaters *2*. There is a separate thermostatic return trap at the outlet of each heater. These are marked *3* on the diagram. The 2 in. main drops to a sediment pocket at *6*, with a pipe riser as a cooling leg and a thermostatic trap at *5*. This trap connects into a drop leg from another thermostatic trap at *4* taken off the top of the horizontal supply main, presumably to remove air from the latter. The return main *7*, 1 in. in diameter, grades back to drop 10 ft and then there are a strainer *8*, a float trap *9*, and a meter *10*, after passing which the hot condensate enters the sewer.

The alternative sketch *B* has the air vent trap *4* of sketch *A* replaced by an atmospheric air vent *11* on top of the 2 in. steam supply main, and the trap *5* connecting directly to the return main.

There seems to be no excuse for the more elaborate scheme *A* for venting and draining. If *9* is a float trap there is no evidence of any means by which the original air within the system can escape, or by which the continually accumulating air can be removed. Sketch *B* is an improvement over *A*, but apparently the air escape valve *11* will be sealed by steam running along the top of the main before it can relieve the air from the drop leg *6*.

Since there is no boiler in the system, there is no particular need of a water seal in this drop leg and so the air escape valve could be in the return main and should be there so as to serve for air escape from the traps *3*, *3*, *3*, on the unit heaters.

If the arrangement of sketch *B* is applied to the piping of sketch *A* and if the air valve *11* is placed at the lower end of the dry return main *7* the plant should operate perfectly since the thermostatic traps *3* and *5* will pass air as well as water.

There is no need of the exaggerated 2 in. drop leg at *6* and the connection to trap *5*, sketch *B*, could just as well be close to the horizontal main, with the bottom of the sediment pocket not more than 18 in. below the bottom of the 2 in. main.

The provision of the strainer *8* is desirable if trap *9* must be used, or possibly to protect the meter, but I can see no need whatever for trap *9* since all connections to return main *7* are made through thermostatic traps *3* and *5*.

Utilize Heat in the Condensate

The condensate from 450 sq ft of equivalent direct radiation, which is the heating capacity, in cold weather will be about 112 lb per hr of water, probably at about 200 F. If this could be cooled to say, 100 F before being released the heat to be saved would be 112 × 100 = 11,200 Btu, equivalent to about 46 sq ft EDR. Taking into consideration the condensate from the piping, one may safely say 50 sq ft EDR could be supplied by the otherwise wasted condensate. It is an easy matter to install a radiator—or better, an additional unit heater *12* —on the inlet side of the meter, as suggested in sketch *C*.

In sketch *C* the return rises into the bottom of the heater, and the water flows upward slowly to overflow out of the top; then the pipe drops and the strainer and the meter are installed. If the air delivered to the econonomizing unit heater *12* is at 70 F it will be easy to cool the condensate to somewhere around 80 F. Water will stand in the drop leg *13* to a height level with the outlet pipe from the unit heater *12*. A thermometer at the inlet to the strainer *8* will permit some observation of the temperature of the departing water.

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*Consulting Engineer. Member of Board of Consulting and Contributing Editors.

If steam costs $1.50 per 1000 lb, and if there is a load factor of 33⅓ per cent, with 240 heating days of 12 hours each, the annual steam consumption of a 50 sq ft EDR heater will be roughly 12,000 lb and the annual saving to be anticipated by the economizer will be $18.

In many central heating systems the water, still at a temperature of perhaps 100 deg, is used via a copper transfer coil to heat the domestic water, or is stored in an insulated tank and is used directly for domestic purposes such as hand washing.

Performance of Coils—

[Continued from page 701]

Table 4—Enthalpy (Total Heat) of Saturated Air[a] (Btu per Lb of Dry Air)

t	h	t	h
25	9.033	55	23.19
26	9.405	56	23.81
27	9.779	57	24.44
28	10.16	58	25.09
29	10.55	59	25.75
30	10.94	60	26.42
31	11.34	61	27.11
32	11.74	62	27.82
33	12.15	63	28.53
34	12.57	64	29.26
35	12.99	65	30.02
36	13.42	66	30.78
37	13.86	67	31.57
38	14.30	68	32.37
39	14.75	69	33.19
40	15.21	70	34.03
41	15.67	71	34.89
42	16.15	72	35.77
43	16.63	73	36.67
44	17.12	74	37.59
45	17.63	75	38.53
46	18.14	76	39.50
47	18.66	77	40.49
48	19.18	78	41.50
49	19.72	79	42.54
50	20.28	80	43.60
51	20.84	81	44.70
52	21.41	82	45.82
53	21.99	83	46.96
54	22.58	84	48.13

[a] Reprinted from "New Tables of the Psychrometric Properties of Air-Vapor Mixtures," HEATING, PIPING AND AIR CONDITIONING, January, 1938, p. 1.

senting the final condition of the air will move down along the condition curve toward the saturation curve.

As stated above the condition curve can be obtained by plotting the points computed by means of Equations 1 and 8. However, there is still another method of plotting the condition curve that is entirely independent of these two equations. This second method is discussed later. The condition curves plotted by these two different methods agree very closely for the usual range of conditions ordinarily encountered. For any given final wet bulb temperature, the final dry bulb computed by means of Equation 8 may sometimes be a fraction of a degree higher than the dry bulb read from the condition curve plotted by the second method to be described later. However, if the initial wet bulb temperature is very high, say about 80 deg, and the dry bulb is very low —that is, the air is almost saturated—the difference appears to be approximately 2 deg. Inasmuch as this latter condition is rarely encountered in actual work, this deviation is not important. For the usual range of operating conditions, the curves obtained by the two independent methods agree very closely, and frequently they coincide.

In order to understand the drawing of the condition curve by means of the second method mentioned above, the meaning of surface temperature and how it is computed must first be understood. This will be explained next month.

[To be continued]

Inspection of Piping—

[Concluded from page 705]

judgment. While it has as yet not been adopted by the A.S.M.E. unfired pressure vessel code or by the pressure piping code, it appears to be receiving very favorable recognition and the author feels there is but little doubt that it will be adopted by those bodies. It is the method that has been used for some time by the United States Navy Department and it is believed that it will become the universal method for the testing of welding operators.

Procedure Must Be Followed

While an adequate procedure of welding is necessary, and competent welding operators cannot be dispensed with, it must be clearly recognized that nothing has been gained by investigating a procedure for welding unless that procedure is followed in the actual construction. It, therefore, seems necessary that in making inspections of fusion welded pipe lines, the inspecting agency must satisfy itself that the procedure for welding is adequate and that the welding operators employed are competent.

The inspecting agency must also be satisfied that the procedure for welding is being followed during actual construction and that only competent welding operators are being used. For a complete inspection, therefore, the inspecting agency must witness the qualification of the procedure and the tests of the individual welding operators. It must also make sufficient inspections during actual construction to ascertain to a reasonable degree of certainty that all requirements essential to the safe operation of the line have been met. This latter requirement means that an inspector must be present during a considerable portion of the time when the work of welding is being carried out.

Storage System Ideal for Churches

The idea of churches being air conditioned by a "giant thermos bottle" was discussed in Boston last month at the meeting of the Power Engineers Association. Walter A. Grant, district chief engineer for Carrier Corp., described this method as being ideally adapted because of the short periods during which churches are used.

"It appears self-evident," he said, "that an application such as air conditioning a church for only a few services a week will permit the use of a very small refrigeration plant storing up cooling effect 24 hours per day for the six week days, and releasing all of this stored cold for the relatively few hours of actual use." Water is gradually cooled and stored in a large insulated tank, and is released to the air conditioning system when needed.

Air Conditioning Overcomes Summer Slump in Bowling

AIR conditioning of bowling alleys has "beyond doubt done more than anything else to eliminate the summer slump that every bowling operator dreads," according to C. J. Butts, manager of the Ritz Bowling Palace, Salt Lake City, Utah. The Ritz alleys have found that many persons who had never bowled during the regular season were attracted by their cool and inviting atmosphere during the hot summer months and became ardent fans. Mr. Butts believes the air conditioning of amusement parlors not only adds to the prestige and identity of the establishment, but is a continuous source of good-will advertising.

By A. A. Maycock*

Air conditioning eliminates the summer slump that every bowling operator dreads, according to C. J. Butts, manager of the Ritz Bowling Palace, where a deep well water system is employed. . . . The cool and inviting atmosphere in summer attracts many persons who have never bowled before, converts them into ardent fans

The Ritz establishment is cooled by a system utilizing deep well water. The building is 180 ft long by 108 ft wide. The front section is divided into a restaurant, which serves also as a thoroughfare to the alleys, a beer parlor, which also serves as a thoroughfare to the alleys, and a store. The rear section of the building, which is 140 ft long, includes the bowling alleys, of which there are 18, as well as dining space contiguous to the restaurant and kitchen, locker rooms for men and women, with adjacent toilets, and a balcony providing additional seating space. Seating space for spectators is also provided on the main floor immediately to the end of the alleys. In the bowling alley section, the ceilings were left open up to the beams, giving a varied ceiling height of 15 ft to 32 ft in the center. The ceilings were insulated.

In the beer parlor, restaurant and store, the ceilings were furred to 14 ft 6 in.

Deep Well Water or Mechanical Refrigeration?

In the initial studies of the proper method to be employed in the conditioning and cooling in the summer, consideration was given to the use of either deep well water or mechanical refrigeration. It was determined that approximately 100 gpm of well water would be required on the basis of 57 deg in order to cool the building from 100 deg to 80 deg. A well was driven and preliminary tests indicated that 100 gpm of water at 52 deg would be obtained; this was regarded as favorable for present as well as for future performance.

It is known that the underground water reservoir at Salt Lake City is regulated largely by a long period of drouth or of a sustained period over several years of heavy rainfall. For several years we have been experiencing drouth. It appears that the cycle has turned and that we now are entering a period of heavy precipitation, under which condition the underground reservoir should build up.

The hourly operation cost for the pump for the delivery of water was estimated at 9c. Were comparable effects received with mechanical refrigeration, the cost would be 69c per hour (44c for electricity and 25c for water for the cooling coils.) It was also estimated that the capital investment necessary for the installation of the heating and cooling, using well water, including the cost of the well and the pump, would be about $13,000. Were mechanical refrigeration used, the extra capital investment required would be about $7,000 over deep well water, with comparable results.

Areas Are Sectionalized

The cooling load for the bowling alleys, restaurant and beer parlor was calculated on the basis of the evening load. During the early evening, the temperature would be as high as during the daytime, but no sun effect would be experienced. As a result, the maximum load of occupancy as far as people, lighting, cooking, etc., were concerned was calculated and the cooling for those sections designed accordingly. The store presented a different problem. Its peak load would occur in the afternoon when there would be the west sun effect and in addition the maximum occupancy, and the load was calculated accordingly for this section.

It was desired to sectionalize the cooling system as far as required, but not to sectionalize more than absolutely necessary in order to keep down the cost of installation. In the State of Utah, beer parlors cannot operate the full number of hours that a bowling alley and restaurant can. As a result, it was desired to sectionalize the beer parlor so this particular part of the building could be closed when it could not be legally open. Inasmuch as the restaurant would operate whenever the bowling alleys were in operation and also due to the fact that it serves as the main thoroughfare to the bowling alleys, the cooling and heating for the restaurant and dining room were made part of the bowling alley section.

Because the load requirements as well as the hours of operation would be considerably different for the store than either of the other two sections, this was made a different section also. Further consideration was given to sectionalizing this part as this section might be operated by the owner or might be leased. If leased, it would lend itself toward the payment of operation expense, such as electricity for fan operation to a charge-back to the store operator.

It was considered desirable to operate under recircula-

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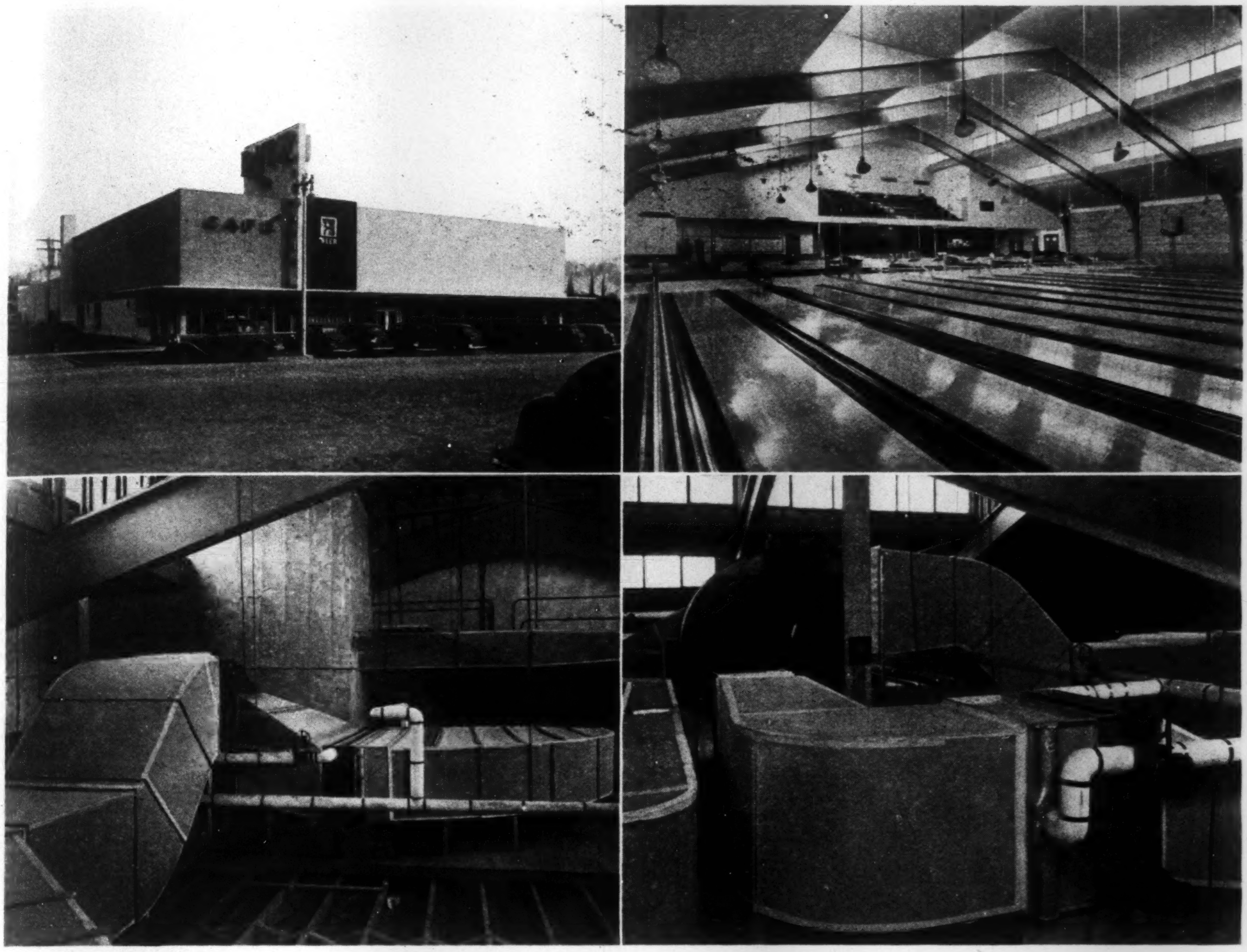
*A. A. Maycock Co.



The Ritz Bowling Palace is cooled by a system using deep well water. These views show the building and some of the air treating equipment. The galvanized sheet metal ducts are insulated with ½ in. board type insulation

tion at all times on the three sections inasmuch as occupancy in relationship to volume is very low. The dissipation of body odors, etc., could be easily absorbed in a building having the cubage given not only by over-all dimension but also by ceiling height. Easier operation was thus achieved, as well as lower cost operation and less expense involved in control. Inasmuch as only about one-third of the floor area in the bowling alley section is occupied, the circulation was isolated as far as possible to these particular points. The air supply is overhead in all instances. In the bowling alleys, the air projects over the seats in the balcony section, with the recirculation through floor registers beneath the seating section in the spectator's area. In the restaurant the air is projected overhead, with returns in the riser of the footstool surrounding the counter. Air is also recirculated at the floor in the dining room section.

In both the beer parlor and the store, the air is projected in the front over the windows and recirculated at the back near the floor level. The kitchen for the restaurant is exhausted with a propeller fan.

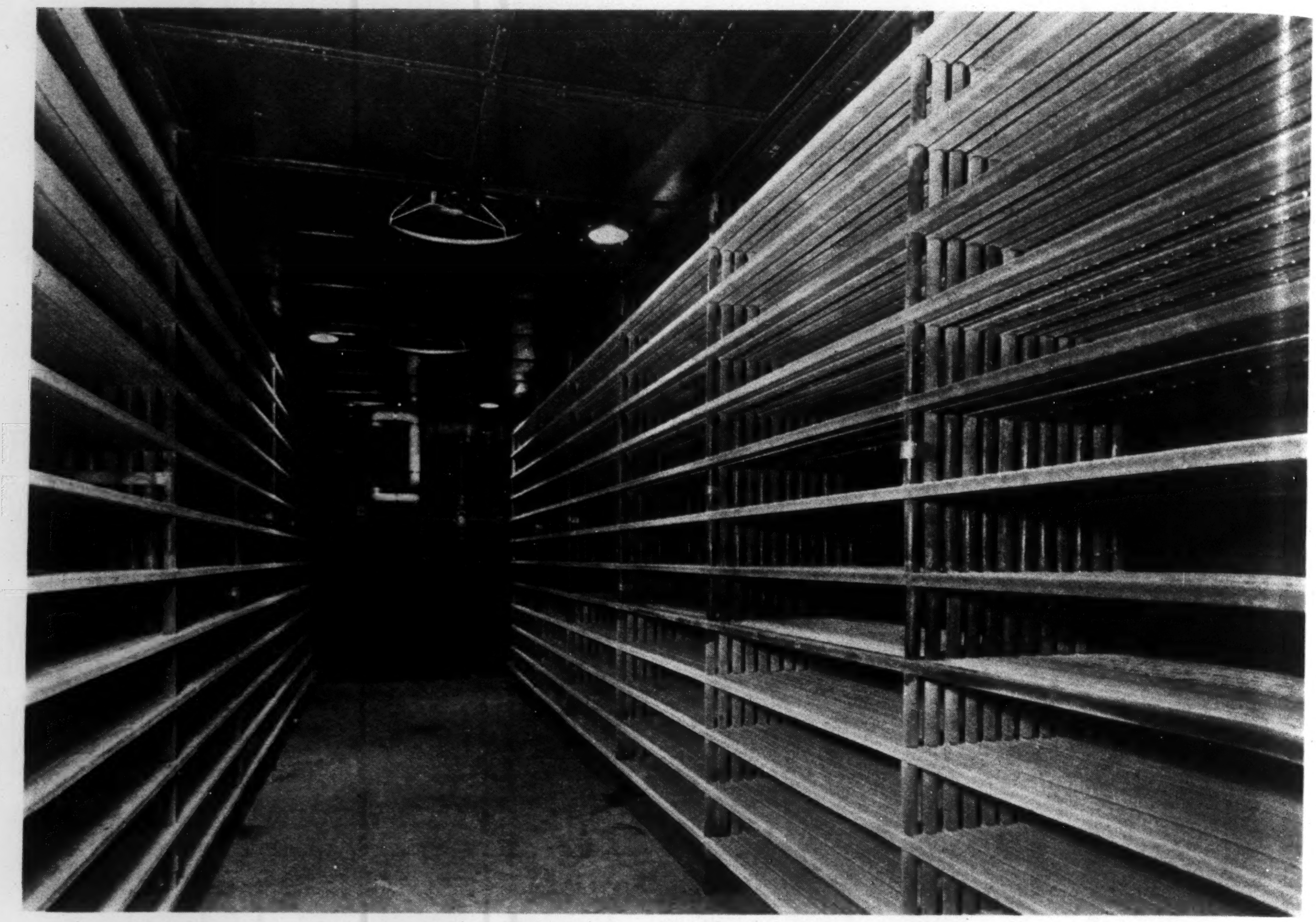
The heating is handled with blast heaters, single coil deep, having adequate capacity to handle the heating in the ventilated spaces. The heating of locker rooms, toilet rooms, etc., is by copper convector radiators, with a two pipe steam system. The cooling is handled by the cold well water circulating through double stacked cold water coils, employing counterflow between the water and the air. The heating and cooling coils were calculated on a velocity of 800 fpm through them.

Automatic heating and cooling is obtained with the use of thermostats and motorized valves regulating the flow of steam in the winter through the blast coils, and water in the summer through the cooling coils; 36,300 cfm of air is delivered to the three spaces with three fans driven through variable pitch v-belt drives.

Steam for heating is provided by a steel boiler for gas firing rated at 21,250 sq ft, which maintains the pressure in the wintertime, and operates from an aquastat in the summertime to give year around hot water from a tank with a submerged coil. The boiler was selected oversize to handle added future load should it develop.

Well water lends itself ideally toward conditioning in this section, as dehumidification is not of great importance, for the mean relative humidity during the period of maximum cooling requirements is approximately 20 per cent.

The architects for the installation and under whose guidance and direction the engineering was done were A. B. Paulson and Lowell E. Parrish. The engineers were the A. A. Maycock Co. and the contractors were the A. H. Walsh Co.



The shelves in this freezing room are constructed of rectangular tubing, forming almost a solid shelf to give excellent contact with the product

Rectangular Tubing Used as Refrigeration Piping

By **William B. Vilter***

[as told to a correspondent]

RECTANGULAR steel tubing is used as piping at the refrigeration plant of H. A. Nieman & Co. at Thiensville, Wis. The plant was installed for the fast freezing of horse meat to be shipped to the company's fox farms for feeding foxes, and has capacity for freezing 50,000 lb of meat in 24 hours. It has also been employed for freezing peas and corn, as the company is also interested in that business.

The installation comprises three cork insulated rooms having two sections of shelf coils 13 pipes high, made up of rectangular tubing. The tubes are fitted closely together and thus form an almost solid shelf which gives excellent contact between the product being frozen and the coils and insures faster freezing, for there is more flat contact surface with the rectangular tubes than if round pipes were used. Another feature of the installation is that the patented method of constructing the coils makes possible perfect flooding without the use of an accumulator.

The rectangular tubing of which the shelves are formed is 2½ in. by 1½ in. in size; 1¼ in. extra heavy nipples welded on are used as spacers between the shelves. Square and rectangular tubing of this kind is made the way steel pipe is, is available in a range of sizes, and has of course been used for many purposes. It has not, however, been used as piping to any great extent and the Nieman installation is believed to be one of the first to use it in this way.

Refrigeration is furnished by a 7½ in. by 7½ in. twin cylinder ammonia compressor and a 10¾ in. by 8½ in. twin cylinder ammonia booster compressor with capacity reduction pockets which make it possible to vary the refrigeration capacity within a range of 50 per cent less than the maximum. Using one compressor or single staging would not have been feasible, according to the engineer, because of the high power requirements, for low suction pressures are necessary to produce the temperatures of —30 to —40 F needed. The ammonia gas is of course evaporated within the rectangular tubing making up the shelves.

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*Vilter Mfg. Co.

As shown in the illustration, propeller fans are installed in a false ceiling to circulate the air in the rooms, and the air circulation speeds up the freezing process. The shelves are about a foot away from the walls of the rooms, and the air circulates from the back of the shelves to the fans and across the trays placed on the shelves.

During the summer of 1938, a quarter of a million pounds of peas were frozen. Nieman prefers peas having a bright green color, running somewhat larger in size than the canning variety. Care is taken to keep the peas cool while in transit to the plant. They are then washed, graded, and blanched for a minute and a half in boiling water or steam at 212 F. When the blanching process is completed, the peas are sprayed with cool water and cooled to about 60 F. For this purpose, a flume is used in which peas and cold water at 50 F are pumped. When this cooling is completed, the peas go through a squirrel cage dryer and are picked over for splits and foreign particles.

The peas are then put in trays with shallow wire screen bottoms, and these trays are placed on the rectangular tubing shelves. After being frozen they are packed in cellophane bags, sealed and placed in cardboard cartons which are stored in rooms held at about zero.

Simonds Saw Occupying Pioneer Windowless Plant

SEVEN years ago, the Simonds Saw & Steel Co. built at Fitchburg, Mass., what is said to be the first windowless factory for American industry. Because of the depression, it was never equipped or occupied, but recently the company decided to complete the plant and centralize its three multi-story fabricating plants, totalling some 17 acres of floor space, in the single five acre room of the new one story building. Original idea in building the windowless plant was to provide working conditions for employees which "would equal those of any executive office," and windowless construction was adopted to aid control of light, temperature, and noise. With these factors under control 24 hours a day, it will be possible for the plant to operate on two or three eight hour shifts under identical working conditions—winter or summer, rain or shine.

With acoustic ceiling overhead and special acoustic tile lining the solid brick walls, the plant will be lighted by combination mercury vapor mazda fixtures 18 ft above the floor to produce shadowless illumination of uniform 20 footcandle intensity or better. Columns are spaced 40 ft apart along eight unobstructed 70 ft aisles in which raw materials, machinery and finished product will be laid out for straight line production in nine individual, self-contained production lines. The major lines include those for wide band saws, crosscut saws, circular saws for wood and metal, machine knives, files and hacksaws, specialty items, inserted point saws, points and shanks, and narrow band saws for wood and metal cutting.

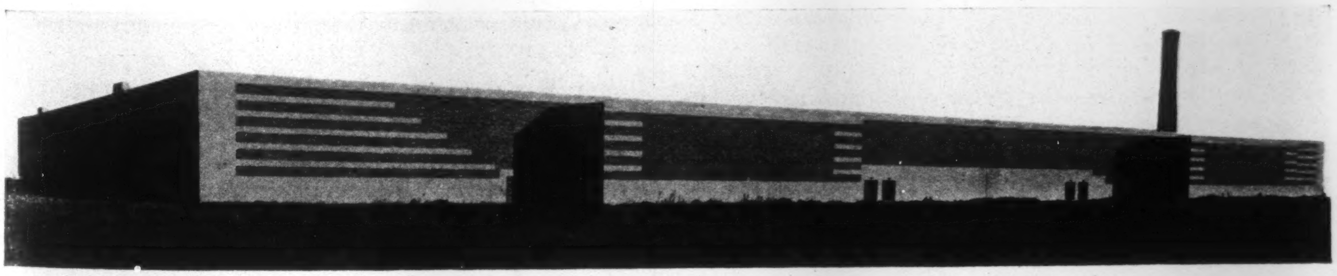
To meet the ventilating and cooling requirements of the plant, where heat treating furnaces, annealing ovens and more than a thousand individual motor driven machines will operate in a single room entirely without partitions, an evaporative cooling system was decided upon. Air will be circulated through the building at a rate of approximately 400,000 cfm, equivalent to six changes per hour. It will come into the structure through louvers on the end walls of four lean-to buildings which adjoin the manufacturing plant at points near the corners on either side of the structure. The air will be blown through water sprays in each of these structures and will be distributed through the building via 3000 lineal feet of ducts which range from a size of 68 in. by 90 in. at the fans to 26 in. by 20 in. at the outlets farthest from the fan units. The ducts are made of copper bearing steel varying in gage from No. 16 to No. 20, depending upon the duct size.

All used air will be exhausted through ventilating hoods in the roof above furnaces and other equipment generating heat, where aprons extending from the ceiling to the bottom chord of roof trusses will trap heat at its source. The air will enter the plant at the wet bulb temperature and it is estimated that the rapid exhaust of air through the roof above the heat generating units will maintain highly satisfactory conditions throughout the plant.

In winter, the air will be washed, heated and humidified. The air handling apparatus in each of the four lean-to buildings serves one quarter section of the manufacturing building.

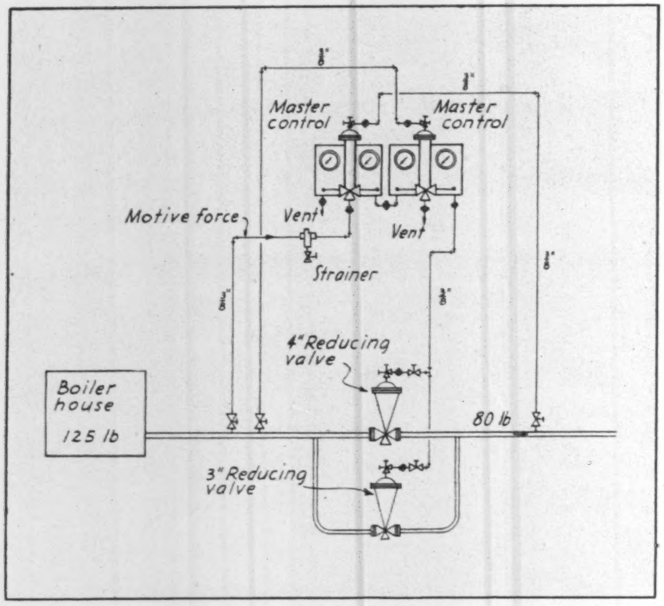
The Austin Co. are the engineers and builders of the plant.

The Simonds plant, built in 1931, will have an evaporative cooling system



Steam Pressure Control with Load Limiting Feature Aids Efficiency

An interview with V. E. Hupp*



The steam reducing control system with load limiting feature. No. 2 master control is to the left, No. 1 master control to right

An advanced method of steam pressure control is employed in the new heating plant in Chicago's Lincoln Park, which furnishes heating steam for the zoo buildings, the floral conservatory, and the maintenance buildings, as well as steam for driving pumps and four turbines. The heating job totals about 110,000 sq ft of radiation.

During the past two years the heating system has been modernized. Worn out boilers and chain grates which had been in service for 30 years were replaced by two modern 400 hp, low head room, high set water tube boilers with spreader type stokers. The boiler room was equipped with mechanical coal and ash handling systems and instruments. The average efficiency of the boiler plant has been increased from about 50 per cent to well over 70 per cent. While the new equipment investment was a substantial one, it should pay for itself within half the useful life of the equipment.

Piping System Renewed

The piping from the boiler house to the various buildings was entirely renewed. With the old system, the pipes had been laid in trenches. Water pipes had been subsequently laid over the steam pipes, making the latter practically inaccessible for repairs. New rectangular tunnels for the steam pipes were built of reinforced and waterproofed concrete 7 in. thick. The main steam tunnel is about 2350 ft long, and is 6 ft, 6 in. high by 5 ft and 4 ft wide, inside measurement. This tunnel is properly graded and the surrounding soil was drained during construction. Part of the tunnel floor is below the level of Lake Michigan, which is less than 200 yards distant at some points. There are also some thousand feet of lateral tunnels, 3 by 3 ft, inside measurement, which serve the various buildings. The steam mains are a maximum of 6 in. in diameter and the return lines a maximum of 3 in., all piping being of wrought iron. The high pressure lines are insulated with double thick 85 per cent magnesia, and the returns and low pressure lines with standard thick magnesia. At five locations in the tunnel system there are twin sump pumps of 25 and 50 gpm capacity to keep the tunnels in a dry condition.

With the old trench piping, whenever it rained, it was necessary to burn additional coal, as the water which filled the pipe trenches played havoc with the steam in the practically unprotected pipes. These heat losses are happily a thing of the past.

The new boilers deliver steam at 125 lb pressure and 75 deg superheat to provide dry steam at the points of distribution. For heating purposes, the steam is reduced in the boiler house to 80 lb pressure, at which it is piped to the points of distribution in the buildings, where further reduction to from 2 to 5 lb is effected for both direct and indirect radiation.

Sudden Temperature Changes a Problem

The heating problem in Lincoln Park presents one difficult angle. The buildings, including a conservatory and propagating houses with large glass exposures, are situated near the lake front where sudden changes in wind direction which occur frequently are followed by

Because 100 lb is the minimum pressure required for operating pumps and other steam driven equipment at the central heating plant for Chicago's Lincoln Park without impairing the service, the reducing valve supplying the heating mains with 80 lb steam is so controlled that when sudden outdoor temperature drops occur, the reducing valve functions as a spill-over or relief valve. Thus, the steam delivered by the boilers is spilled into the 80 lb mains until enough time has elapsed for the boiler output to increase sufficiently to supply the full 80 lb steam demand. The valve then reverts to operation as a reducing valve, while the boiler pressure gradually rises to its normal of 125 lb, without waste of coal

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*Heating Engineer, Chicago Park District.

drops in temperature of 15 to 20 deg within a short period. These changes put rapidly increased demands on the heating system and there is a tendency for the boiler pressure to drop sharply. With the old plant, it was impossible to combat this condition successfully, and adequate heat could not be provided. Now, when the wind changes and the temperature drops, the heating system automatically adjusts itself to the increased demand, and without excessive firing of coal.

Minimum Boiler Pressure 100 Lb

In the boiler house are two master steam pressure controls, so connected that, together, they operate either or both of two reducing valves, the smaller valve being used in mild weather and the larger in winter, according to the load. These valves, situated at the entrance of the main steam tunnel, cut down the pressure of the steam delivered by the boilers from boiler pressure to 80 lb as long as the boiler pressure is 100 lb or more. This 100 lb pressure is the minimum required for operating the pumps and other steam driven equipment in the plant, and it is imperative that the boiler pressure does not drop below this figure.

When the heating steam demand increases suddenly, following a sharp temperature drop, the boiler pressure may reduce to 100 lb. If this reduction occurs, the reducing valve regulated by the master controls automatically becomes a spill-over or relief valve. The steam delivered by the boilers is spilled into the 80 lb header until enough time has elapsed for the steam output to increase sufficiently to supply the 80 lb demand. The valve then reverts to operation as a reducing valve, maintaining the desired 80 lb pressure for delivery to the steam main, while the boiler pressure gradually rises to its normal of 125 lb, without waste of coal from too rapid firing.

The master controls thus regulate the reducing valve not only to maintain the 80 lb pressure desired in the steam lines but also to prevent the boiler pressure from being reduced below the requisite 100 lb.

The controls perform their function in the following manner:—The tops of the diaphragms of the two master control valves are connected by piping to the steam main, No. 1 to the high pressure side and No. 2 to the low pressure side of the reducing valve. Adjustable springs control the pressure applied to the bottom side of the diaphragms.

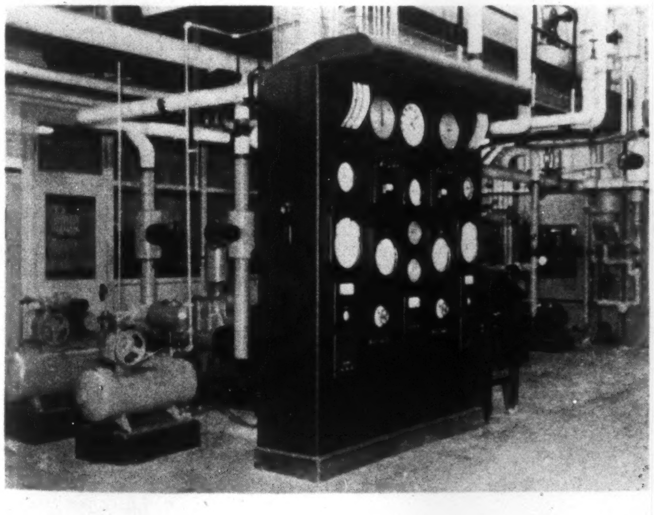
When the supply steam pressure rises above 100 lb, this pressure overcomes the pressure of the spring of No. 1 control valve, lowers the valve stem and permits steam pressure to be applied to the top of the diaphragm of the reducing valve through a valve situated at the lower end of the master control, which pressure opens the reducing valve. When supply steam pressure drops to 100 lb, the pressure on the diaphragm of the master control is overcome by the pressure exerted by the spring forcing the valve stem up, which in turn closes off the pressure admitted to the reducing valve diaphragm, vents the reducing valve diaphragm to atmosphere, and permits the reducing valve to close.

When the reducing valve is closed, the pressure on the low pressure side of this valve reduces, thereby decreasing the pressure on the top of the diaphragm of No. 2 control. The adjustable spring beneath the diaphragm of this control raises the valve stem which opens the valve to admit steam through the line from the high pressure side of the reducing valve to No. 1 control valve. This pressure will not pass No. 1 control valve to open the reducing valve until the pressure on the high pressure side of the reducing valve is above 100 lb.

When boiler pressure rises above 100 lb, the valve of No. 2 control opens to admit the pressure supplied by No. 1 control to the reducing valve diaphragm which opens the reducing valve and satisfies the demand on the low pressure main. When this demand for pressure is satisfied the pressure on the top of the diaphragm of No. 2 control overcomes the spring pressure, lowering the valve stem. This in turn closes the valve, cutting off the steam supply to the diaphragm of the reducing valve and venting this diaphragm to atmosphere, which permits the reducing valve to close by spring action.

The modernization of the heating system was designed and the construction was supervised by the engineering forces of the Park District, under the immediate supervision of Mr. Hupp and the general direction of W. I. Bell, assistant general superintendent.

Modern instruments were installed



Check Charts for Air Conditioning Calculations

By J. M. Dean*

ON the next page appear Charts Nos. 11 and 12 of this series, the first of which was published in the May, 1938, H. P. & A. C. with a general introduction. These charts were developed in connection with the author's duties as air conditioning engineer for an electric power company in an endeavor to reduce the work of checking air conditioning jobs to a simple routine procedure.

Chart No. 11 is for computing pounds per minute of ventilation air, recirculated air, and total air with the cfm and cu ft per lb of ventilation air and recirculated air known. Chart No. 12 is used for determining conditions of mixtures of two quantities of air.

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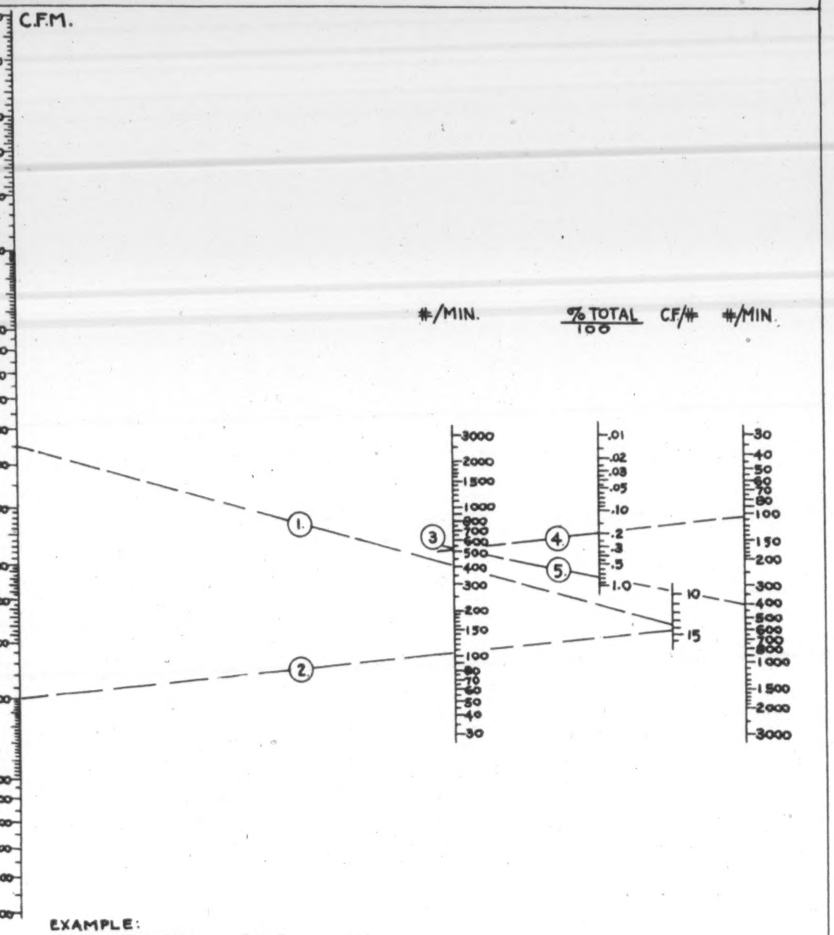
*Air Conditioning Engineer, Des Moines Electric Light Co. All rights reserved.

Check Charts for Air Conditioning Calculations

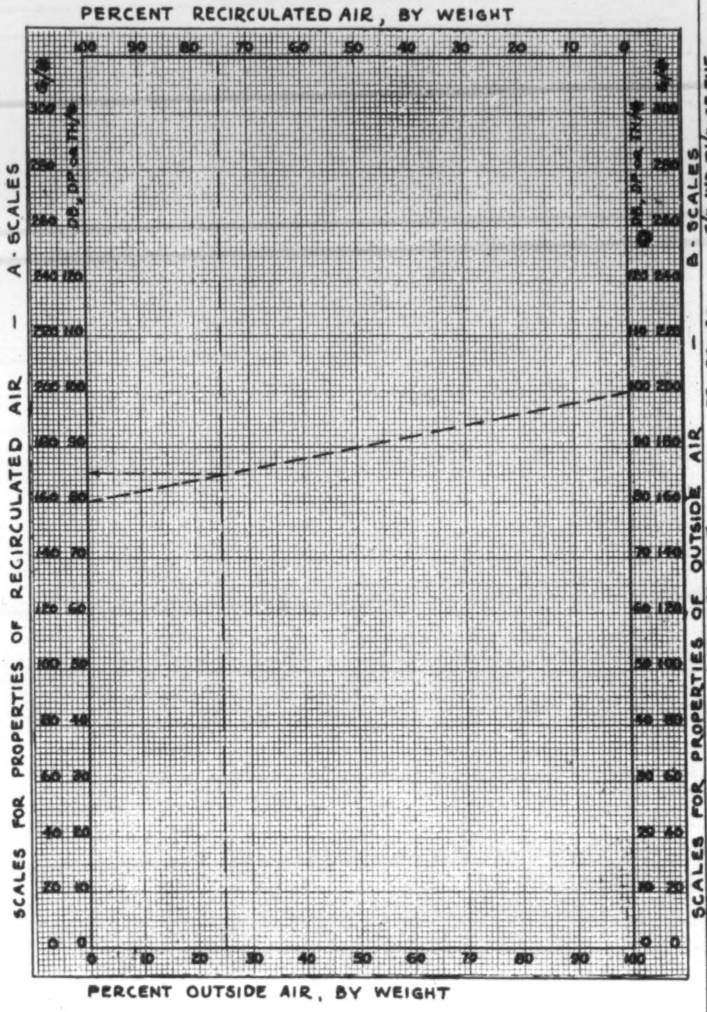
[See preceding page]

Chart No. 11



EXAMPLE:
7000 CFM TOTAL 510# 100%
1500 CFM VENT. AIR 105# 79.5
5500 CFM RET. AIR 405# 20.5
1. CONNECT 5500 CFM & CF/# INSIDE AIR; READ 405 #/MIN.
2. CONNECT 1500 CFM & CF/# OUTSIDE AIR; READ 105 #/MIN.
3. ADD ABOVE TWO VALUES FOR TOTAL AIR WT., 510 #/MIN.
4. CONNECT 510 #/MIN. & 105 #/MIN.; READ .205
5. CONNECT 510 #/MIN. & 405 #/MIN.; READ .795
ADD THESE TWO VALUES, TO CHECK 1.000

Chart No. 12



EXAMPLE: RA 80 DEG., 75%; OA 100 DEG., 25% .75 × 80 = 60
CONNECT 80 ON A-SCALE WITH 100 ON B-SCALE; .25 × 100 = 25
FROM INTERSECTION OF THIS LINE WITH THE Tm = 85
75% RA-25% OA VERTICAL LINE, PROJECT
HORIZONTALLY TO EITHER A- OR B-SCALE, READING Tm = 85
G/# AND TH/# OF THE MIXTURE CAN BE SIMILARLY DETERMINED. ALSO, THIS CHART CAN BE USED FOR OTHER THAN RA & OA MIXTURES

Applying Control to Hotel Heating

By O. E. Stossel*

DEMAND for economy and better service in transient and residential hotels has led to the installation of a great variety of heating controls, most of which are excellent pieces of engineering work and have been very successful with different types of heating systems, especially in smaller buildings. If there have been failures recorded in the case of some large hotels, they have been caused usually by trying to make the control do something that was beyond its scope.

The load curves of large hotels are complex and differ in each individual case. The larger and more elaborate a building is, the more factors there are to be taken into consideration. It is unfortunate when a good heating control is discredited because the need for a survey of conditions within the building has not been recognized.

Conditions Influencing Heat Demand

It can be easily understood that the heat demand of a building may be influenced by conditions which are most difficult for a mechanical device to detect. In severely cold weather, for instance, the room temperature must be kept higher to compensate for the increased radiation loss of the human body to the cold walls and windows. There are many other instances in which more or less heat should be given than outside conditions alone might warrant. Especially in large buildings, it is our experience that the most satisfactory results will be obtained if the attention of an experienced operator can be combined with a control that will provide efficient heating under all conditions.

The installation of radiator orifices is a valuable aid to better performance of a controlled heating system. One should guard carefully, however, against the habit of maintenance men who like to remove orifices from radiators for which they have had complaints. If this tampering has been done many times, the system may be worse off than before installation of the orifices. Whenever the system is in poor condition and afflicted with many trapped lines which cannot be eliminated, it is advisable to operate with a high pressure differential and small orifices, in order to maintain circulation. For greater economy, however, the pressure in the feed lines should be kept as low as the condition of the system will permit.

Condensation in the feed lines should be kept at a minimum. The heat loss in the feed lines is, of course, determined by the difference in temperature between pipe and surroundings and by the quality and thickness of the insulation. Sub-atmospheric steam, being lower in temperature, will diminish the heat loss somewhat. There is also a second and perhaps greater loss in the feed lines due to the condensation which is carried back directly to the vacuum pump without ever getting into the radiator. The difference in temperature between the condensation from the feed line traps and the condensation from the radiators may be as high as 80 F.

In most cases excessive condensation in the feed lines can be avoided. The steam should be reduced sufficiently to maintain from the reducing valve up to the radiator never less than 5 F superheat. The greater part of the condensation, in this way, will be re-evaporated before it is discharged through the riser traps. A thermometer and a compound gage connected to the last radiator valve will tell if the steam there is still in a condition to re-evaporate the condensate.

In a plant with high boiler pressure the conditions are still more advantageous if one takes care to carry a sub-atmospheric pressure from the reducing valve on. In a system that has a tendency to carry excessive condensation it may be advisable to increase the boiler pressure a few pounds. In many plants, of course, exhaust steam from pumps and engines represents a large percentage of the total heating steam, which makes it difficult to prevent excessive condensation in the heating lines.

Installing Control on Savings Basis

A word of warning should be spoken in connection with heating controls installed on a savings basis. Steps must be taken beforehand to enable one later to determine the exact amount of the savings. Few hotels have sufficient and good metering equipment. Without this equipment, an increase or decrease in other than the heating load or changes in boiler efficiency cannot be accurately checked, even if the owner cooperates 100 per cent. Unless one is sure that the building is not under-heated at various times, one should be very cautious with promises of savings. An improvement in service only, achieved by the control, cannot be expressed in dollars and cents.

A quick check-up on a heating plant is made very easily. The heating plant can be considered to operate efficiently if the temperature in the building is right and the temperature of the returned condensate low. The correct return temperatures differ for each plant, and increase inversely with the outside temperature. But even in very cold weather the condensate temperature in a vacuum system should not rise over 130 F.

If there are never any complaints from tenants, one can be fairly sure that the building is being over-heated.

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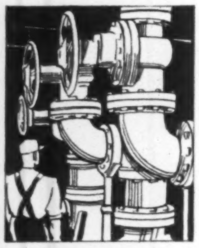
*The author is connected with a hotel management concern.

How to Install and Operate

Deep Well Turbine Pumps

THE following paragraphs have been taken from the deep well turbine pump section of the "Standards" of the Hydraulic Institute:

When installing deep well turbine pumps, care should be taken to see that all parts are properly cleaned; and when handling, that the shaft, outer pipe, and shaft enclosing pipe or tube, when used, are not bent; then follow carefully the detailed instructions furnished by the manufacturer.



Never force the pump into the well. It must slide into the well freely and easily. Forcing or cramming the pump into the well may cause serious damage; in fact, it may be impossible to start a pump that has been forced into a crooked well. If the pump is forced, the discharge column and the lineshaft may be bent, which would result in binding the lineshaft in the bearings so that satisfactory operation would be impossible. Deep well turbine pumps must hang free in the well when in position, and the head or base plate is grouted into place after being properly aligned.

If pump will not start, check the following:

(a) Low voltage.
(b) One or more fuses burned out. (Test voltage on all phases of motor terminals).
(c) Overload relay tripped.
(d) Motor defective.
(e) Starting equipment defective.
(f) Pump sand locked. (Try loosening with Stillson wrench).
(g) Crooked well.
(h) Packing too tight. (If used).
(i) Impellers improperly adjusted, rubbing on top or bottom.

If pump will not deliver water:

(a) Lowest impeller not submerged.
(b) Pump speed too low. (May be due to low voltage or frequency).
(c) Pumping head too great.
(d) Suction pipe or impellers clogged up.
(e) Suction strainer clogged up.
(f) Lineshaft broken.
(g) Pump operating in wrong direction.
(h) Discharge valve closed.

If pump will not deliver sufficient water:

(a) Air or gas in water.
(b) Leaky suction pipe. (If the pumping water level is below lowest impeller).
(c) Pump speed too low. (May be due to low voltage or low frequency).
(d) Pumping head too great.
(e) Suction pipe or impellers partially clogged up.
(f) Suction strainer partially clogged up.
(g) Discharge valve partially closed.
(h) Excessive drawdown.
(i) Suction lift on impellers.
(j) Worn impellers.
(k) Leak in discharge pipe.
(l) Pressure gage incorrect.
(m) Capacity measuring device incorrect.
(n) Axial adjustment of impellers or rotating element incorrect.

If pump takes too much horsepower:

(a) Too heavy oil being used for lubricating lineshaft bearings.
(b) Sand in water.
(c) Crooked well throwing lineshaft bearings out of line.
(d) Speed too high. (May be due to high voltage or frequency).
(e) Discharge head different from that for which pump was designed.
(f) Crooked lineshaft.
(g) Vibration.
(h) Packing too tight (when used).
(i) Impellers improperly adjusted, rubbing on top or bottom.
(j) Improper alignment at pump head.

If pump vibrates:

(a) Worn bearings.
(b) Bent lineshaft.
(c) Motor out of balance. (Get in touch with motor manufacturer).
(d) Crooked well.
(e) Air or gas in water.
(f) Motor shaft not lined up with pump shaft.
(g) Motor coupling out of balance.
(h) Impellers out of balance due to foreign matter caught in them.
(i) Pump head not properly aligned.

If there is excessive wear:

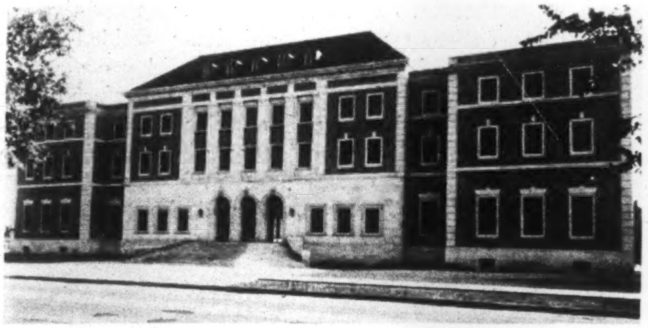
(a) Bent lineshaft.
(b) Improper lubrication.
(c) Crooked well.
(d) Sand in water being pumped.
(e) Vibration.
(f) Misalignment.
(g) Cross threads.

For oil lubricated, enclosed lineshaft pumps, be sure that the oil used for the lineshaft bearings is extremely light. The viscosity should not be over 100 to 150 seconds Saybolt Universal at the well water temperature

Waste Steam Air Conditions New University of Nebraska Building

By Conrad V. Carlson

PUTTING a waste product into productive use is always an interesting subject, for such projects usually involve ingenious methods and careful planning to attain the desired end. Such is the case with the new Student Activities building at the University of Nebraska, Lincoln, where exhaust steam (otherwise wasted) from the turbines is used to produce refrigeration for air conditioning.



Air conditioned Student Activities building

Heat and power for the Activities building, the other main campus buildings, and the State Capitol building are furnished by the University's power plant. The 35 lb exhaust steam from the turbines is used for heating the campus buildings in winter, but in summer has had to be condensed. Steam is supplied to the turbines at 275 lb from the boilers, there being two turbines of 1250 kw rating each. Steam for heating the Capitol is supplied at 275 lb pressure. In summer approximately 10,000 lb per hr of steam is available. The power plant is connected with the campus buildings and the Capitol building with tunnels so that additional piping (such as chilled water) can be run to any building from the power plant.

Although the original Student Activities building plans were to include a complete air conditioning system with the exception of the refrigeration, it was found in the fall of 1937 that finances would permit the addition of the refrigeration unit, so its location became one of the first problems. Making use of the exhaust or waste steam seemed the logical procedure rather than generating electricity, which in addition to incurring losses in transforming the steam to electricity would mean making additional energy at the power plant. The exhaust steam from the turbines for producing the refrigeration may be considered as costing nothing, as, otherwise, it would have to be condensed. The question then resolved itself into that of the location for the steam jet machine at the Activities building or the power plant. Location at the Activities building would have made necessary a cooling tower there. Location at the power plant would not necessitate additional overhead for supervision as the regular operators could take care of the plant in addition to their regular duties. They would also be on duty day and night, an advantage since conditioning is necessary on some evenings quite late in the ballroom. Location at the power plant also permitted using the existing cooling tower.

The principal obstacle to putting the steam jet unit at the power plant was the cost of the chilled water piping and insulation, particularly the latter. The temperature in the pipe tunnel ranges from 115 to 130 F, so with piping carrying chilled water, the insulation must be effective as the chilled water must be piped a distance of 2680 ft. The effectiveness of the insulation is attributed mainly to the care used in applying it. The insulation consists of two ¾ in. layers of wool felt placed with the joints staggered; every joint was carefully sealed with asbestos. An 8 oz canvas jacket was then applied and this was given a coat of special sizing followed by a coat of a special black paint.

The steam jet machine (rated capacity 200 tons) was installed in the north-east corner of the power plant on the ground floor, a little excavating for the chilled water pump being necessary to obtain the required head room. An indication of the care used in making the steam jet installation is that it has been possible to produce such a vacuum that 33 F chilled water has been obtained.

As the cooling load of the Activities building is approximately 125 tons, the machine has additional capacity, it being necessary to operate two and sometimes only one of the four steam jet boosters.

A 50 hp, single stage, motor driven pump circulates the chilled water through the 2680 ft of piping to the Activities building at an initial pressure of 112 lb and a return line pressure of 60 lb. Two 40 hp motor driven pumps circulate the condenser water, only one of which is necessary at the present partial loading.

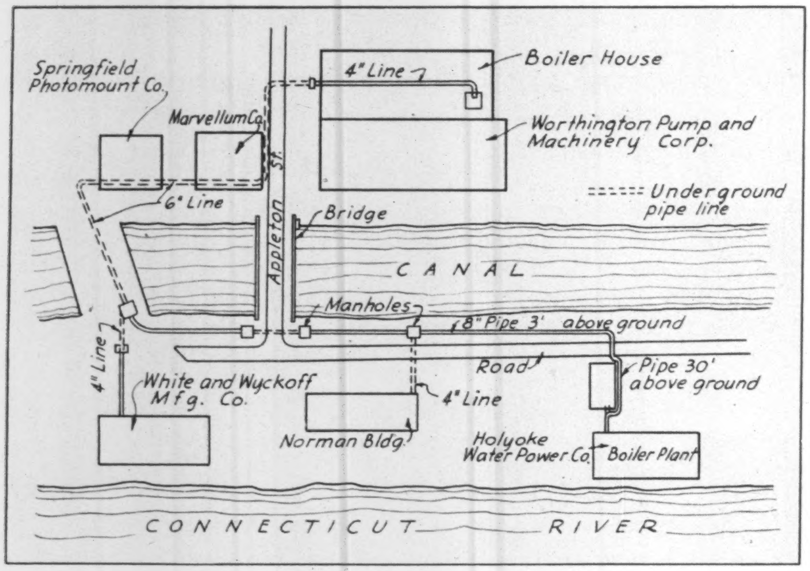
Of the 10,000 lb per hr of exhaust steam available in summer, 3000 lb is used for the steam jet machine, so additional capacity for future air conditioning is available besides the excess capacity of approximately 75 tons in the present system.

Acknowledgment is made to Jacob Schmall, plant superintendent, for the data and information given here. Architects for the building were Davis & Wilson, and William L. Cassell was the consulting engineer. The piping was installed by Newberg & Bookstrom, who also installed the steam jet equipment under the supervision of the consulting engineer and Natkin & Co.

The author was not connected with this installation other than in the preparation of this article.

Holyoke Water Power Company Installs Industrial Steam Line

A welded steam distribution system for supplying steam for process and heating to several industrial customers was recently completed by the Holyoke (Mass.) Water Power Co. The piping is briefly described here, with information on the tests of the welders and the welds and putting the piping in operation



Route of the new steam distribution system

A WELDED piping system, designed to carry steam at 200 lb pressure and 540 F temperature, was recently completed by the Holyoke Water Power Co., Holyoke, Mass., to deliver steam to several large manufacturing plants where it is reduced to pressures suitable for heating and process uses.

The line extends from a boiler plant on the shore of the Connecticut River (see sketch), crosses a railroad right of way on a box truss to a building, continuing along the outside wall of this building to another box truss which carries it across a road and another railroad right of way, turns at right angles and parallels the roadway to a point opposite the Norman Building, with which it connects through a 4 in. line underground.

The first section of the line from the boiler plant to the southerly side of Appleton St. is 8 in. pipe. At this point the main changes to a 6 in. line running overground on concrete piers to a manhole where a 4 in. branch line runs to the White & Wyckoff Mfg. Co. The 6 in. main continues underground to the Springfield Photomount Co. which is served with steam by a 2 in. branch. Continuing underground and through the basement of the Marvellum Co., which is served with steam for heating and process use, the 6 in. main continues underground to property of the Worthington Pump & Machinery Corp. Here the size reduces to 4 in. and the line rises from underground to overhead construction running along the building wall to the boiler house.

Welding Most Important Problem

The welding work was considered to be the most important construction problem. The line is of all welded construction using flanged valves with welding neck flanges at the connections thereto. The specifications required that the entire system should have welded joints made by the electric arc weld process suitable for 300 lb WSP and performed by welders holding Hartford certificates or giving equivalent evidence of their experience and qualifications. Acetylene gas welding was later used as well and about 25 per cent was performed by this process.

All welders were given a stringent test before being employed. Each prospective applicant was required to make a 6 in. pipe weld in position (non-rolling). The weld was then burned by the torch into 1 in. wide sections and each sample thus obtained clamped in a vise close to the weld and hammered backward until it was doubled back upon itself. Any crack showing in the weld meant that the applicant did not qualify.

The welded joints were made without the use of backing-up rings. The ends of the pipe were beveled, the joints being the single 60 deg vee butt type. Arc welding of the joints was done in two passes, the first with $\frac{5}{32}$ in. electrode and the second with $\frac{3}{16}$ in. electrode. An average of 14 joints were welded in a 10 hr day; this included cutting, lining up, tack welding and finish welding.

All welded joints were subjected to a 4 lb hammer test with the line under hydrostatic pressure. The specifications called for a hydrostatic test of 400 lb gage pressure to test the welds and anchors. The main line was tested in five sections as the work progressed. The only unusual incidents occurring during the tests were the failure of a concrete anchor that had insufficient reinforcement, the appearance of a slight crack in a 16 in. brick wall laid in cement mortar which was caused by a temporary brace, and an observed movement of an anchor pier of ⅜ in. caused by the "take-up" of the dead man. For certain branches it was necessary to test a cold main while the adjacent main was hot. In such cases it was necessary to avoid having the valve disc subjected to cold water on one side and steam on the other with resultant warping stresses. To accomplish this a temporary blank steel plate was inserted at the gasket of the cold flange.

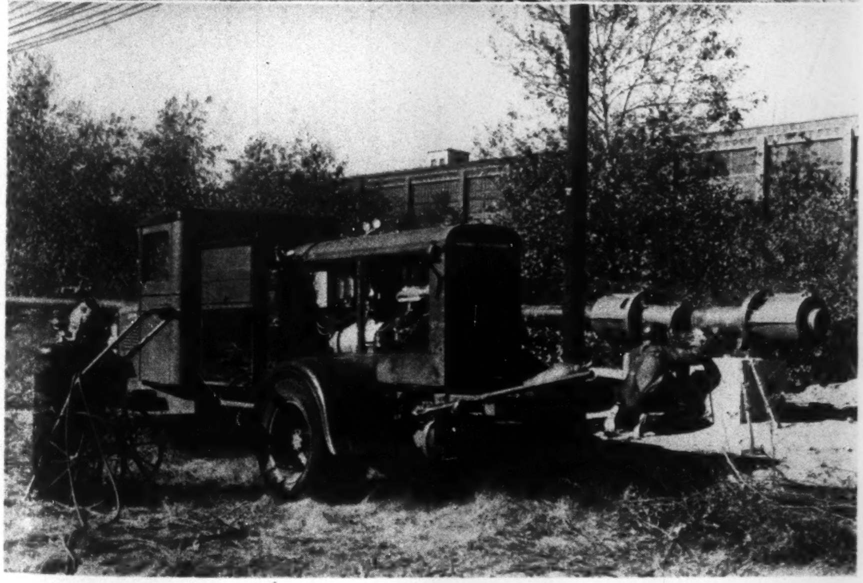
A complete discussion of this industrial steam line, from which most of the information given here has been taken, by E. H. Cameron and Ralph E. Day, was published in the *Bulletin* of the National District Heating Association, Vol. 23, No. 4. Illustrations used here courtesy The Lincoln Electric Co.

Underground Construction

The only exception to the standard concrete envelope underground construction occurs at a 90 ft section located in a passageway between two manholes. This location, in a 4 in. branch line, presented a difficult construction problem. The pipe runs in front of and parallel to a busy shipping platform. Previous exploration had revealed a maze of water, gas and steam pipes and conduits containing underground wiring. Further complications occurred in the form of two 600 volt underground power cables which crossed and recrossed the proposed line making it necessary to dig up a large area and relocate these cables. At the high spot there remained only 20 in. of available space between the top of a conduit containing electric wiring and the surface of the paving.

Standard concrete envelope construction being impossible in this location, the 4 in. pipe was insulated and inserted in a 10 in. pipe. The trench was then excavated, the pipe in its protective sleeve placed in two sections and the 4 in. and the 10 in. pipes welded at the high point. The assembly was then set accurately as to grade and slope and encased in concrete. Such construction proved to be very expensive and is only warranted by similar unusual conditions.

Above—**Welding a dirt pocket and drip line into the piping.** ***Below***—**Arc welding an expansion joint**



Starting Up and Operation

The operation of the line is under the direction of the operating staff of the Holyoke Water Power Co.'s No. 1 generating station. The staff has a splendid record of freedom from interruption of electric service and these men are applying the same careful standards to the steam line operation. On November 12, 1937, steam was first turned on by means of a ¾ in. bypass in the outgoing main valve. The line from the power house to manhole 3, a distance of 2000 ft, was given two hours to warm up before the main valve was fully opened. Each slip joint was closely watched and drip traps carefully observed. From manhole 3 on, the line was then turned on section by section to the end, four hours elapsing between the initial opening of the station outgoing valve bypass and the delivery of steam to the most remote customer.

At this stage, despite the dirt pocket at the drip traps, considerable trouble was experienced from dirt, pipe scale and welding chips and for several weeks the closest of inspection of traps was required. The only modification to original methods that the engineers would make for the next job would be to supply two free blow valves in series, instead of one, at each drip, mainly as protection during the early stages of operation until the line becomes wholly clean.

At the time of starting there was still considerable uncovered pipe and trap discharges were excessive. At present under normal outgoing temperatures and pressures, which are approximately 160 lb gage and 460 F, the discharges are negligible and it is probable that an average interval of 20 minutes occurs between each discharge. During the first week of operation the entire line was patrolled day and night every two hours. At present the line is patrolled every eight hours.

Analysis of Sunshine—Bulletin 911 of the General Electric Research Laboratory, Schenectady, N. Y., is entitled "A Time Analysis of Sunshine," by Frank Benford and John E. Bock, and comprises a paper, presented before the Illuminating Engineering Society. While the study pertains particularly to the natural lighting of buildings, it is of interest to air conditioning engineers as an aid in determining the amount of solar energy received by a room.

Safety Engineers Discuss Mine Ventilation and Heating

HEATING of air that is forced into a mine to keep ice from blocking the ventilation shaft when the temperature drops on the top side was discussed before the mining section of the National Safety Congress last month by R. D. Satterley, superintendent, Inland Steel Co., Ishpeming, Mich.

When the ice blockade problem first arose in the Morris mine, Mr. Satterley related, engineers decided to use the regular pressure ventilating fan and an exhaust fan alternately, so that warm air drawn from the mine would melt the ice formed when the pressure fan pushed in great volumes of cold air. This plan was not successful, however, because of the longer periods necessary to operate the exhaust, which fouled the mine air with gases from decaying timbers in an old part of the diggings.

Detailed descriptions, both of the mine layout and of the heating system which solved the problem, were given by the speaker, who gave data as to costs of construction and maintenance and the amount of time required for supervision of the automatic machinery.

Automatic stokers, thermostats, aquastats and heaters which make up the heating unit added to the ventilation system require a minimum of attention (about two hours a day), he said. Coal feeds by gravity into the stoker from a hopper, which holds a carload of fuel.

When the temperature of the outside air drops near the freezing point, the first of four unit heaters is turned on by a thermostat. A second unit kicks on intermittently at temperatures from 20 deg down to 5, a third comes in between 5 and 15 deg below zero and all four are operating when outside temperature drops to minus 25 deg.

Nominal costs for the installation and operation were quoted by Mr. Satterley, who stated that the installation has entirely eliminated the frequent interruptions of former winters that resulted from ice forming in the ventilation shaft.

Mine Ventilation

Mine ventilation and some of the new developments in this branch of industry were explained before the same section by Raymond Mancha, manager of the ventilation division of the Jeffrey Mfg. Co., Columbus, Ohio.

The principal point emphasized in Mr. Mancha's paper was the high efficiency and low cost of mine ventilation through the use of propeller type fans installed at the bottom of air shafts, compared to the method of exhaust fans operated at the surface. Retention of the surface exhaust fans for emergency use was recommended, however.

Rock Dust Removal

Rock dust removal is of primary importance in maintaining healthy conditions in any mine, but when the diggings are extensive, an elaborate and complicated system of ventilation is needed to solve the problem. Such an extensive system, as used by the Montreal Mining Co., was described by C. M. Fellman, safety and ventilation engineer of the Montreal Mining Co., Montreal, Wis.

With normal production averaging 4000 tons of hematite ore per day, coming from a network of levels and cross cuts at an average depth of 2300 ft, ventilation is an important problem in the Montreal mine, requiring four high powered pressure fans in the main shafts and numerous auxiliary fans throughout the system.

"The necessity of adequate ventilation and the benefits which are derived therefrom are so well known and recognized by mining men that there is no need to dwell upon the subject," declared Mr. Fellman. "Suffice it to say that the ventilation systems of iron ore mines should aim to provide moderate working temperatures, lower the relative humidity, remove heat caused by exposed gob or timber mat, provide fresh air low in dust content to all parts of the mine, and rapidly remove dust, smoke and gases caused by blasting without contacting the workmen."

Because of the extensive underground reaches of the Montreal claim, the ventilation system is divided into 300 ft blocks, each block having its fresh air source and return air outlet, which prevents the passage of dust and contamination from mining operations through the entire territory.

The air currents on main haulage levels are controlled by automatic ventilation doors, which allow passage of motor trains without breaking the main air current.

Auxiliary fans are used in mining and development places which cannot be ventilated by the fresh air currents. The fans are placed in the fresh air currents and the air carried into the working places through a 12 or 16 in. galvanized metal pipe or tube.

Rock headings are ventilated by 15 hp high speed pressure fans, set in fresh air currents with 16 in. tubes to conduct the air to within 20 ft of the breast. The return air current from the breast passes through a pair of water curtains to settle the dust. They are also effective in settling smoke and gases after blasting. During drilling the miners wear air line respirators, supplied with compressed air from underground air lines which passes through a filter which reduces the pressure and heats and humidifies the air if necessary. All drilling is done wet.

Dust counts, as recommended by the U. S. Public Health Service, are made continuously from every part of the mine, providing a constant check on air conditions underground and pointing the way to improvements in the ventilation system.

By R. W. Clark*

This paper—which was presented at the annual meeting of the American Welding Society last month—reports the results of two series of tests on arc welded joints in carbon-molybdenum piping. The first series was made to determine the effect of groove design, whether or not general chipping of each bead is necessary, and the value of a recording voltmeter as a means of checking soundness of welded joints. . . . The second series of tests was made to determine the residual stresses in pipe joints welded at three different temperatures when tested in the as welded, furnace stress relieved, and local stress relieved condition

Welded Joints in Carbon-Molybdenum Piping

THE comparatively recent increase of steam temperatures and pressures in modern power stations has placed much more rigid requirements on joints in piping for this service.

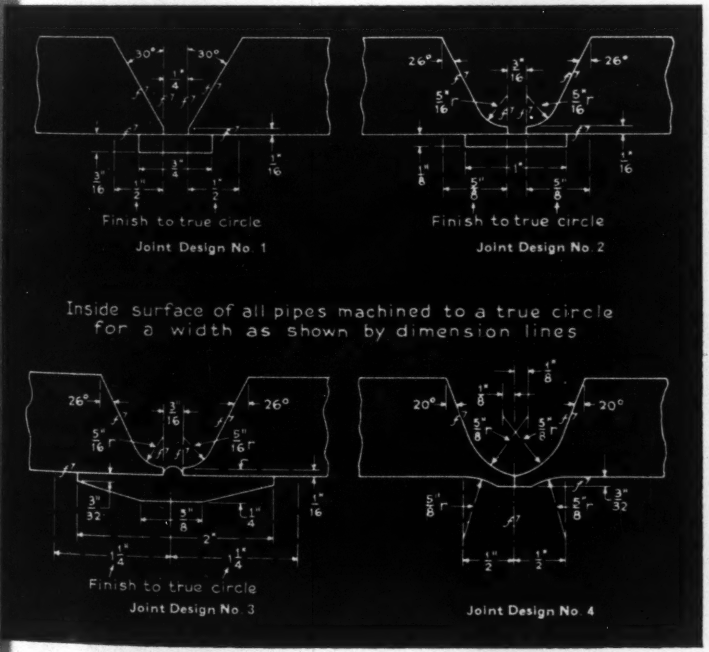
Carbon-molybdenum steel, because of its good physical properties and resistance to creep at high temperature, has been generally accepted as the base material for the pipe, and welding as the most satisfactory method of joining it. However, there are differences of opinion regarding groove design, detailed welding procedure, methods of inspection and heat treatment subsequent to welding of joints for this service.

The first series of tests was made to determine:

1. The effect of groove design, by comparing four representative types.
2. Whether or not general chipping of each bead is necessary.
3. The value of a recording voltmeter as a means of checking soundness of welded joints.

The second series was made to determine the residual stresses in pipe joints welded at three different temperatures when tested in the "as welded," "furnace stress relieved" and "local stress relieved" condition.

Fig. 1—Four types of grooves compared



First Series of Tests

Before starting this series of tests, preliminary samples in both plate and pipe were made to establish the welding procedure for various types of grooves in pipe in the horizontal and vertical positions.

All welded pipe samples were taken from one length of $7\frac{5}{16}$ in. ID and $8\frac{5}{8}$ in. OD seamless tubing, its composition being 0.13 per cent carbon, 0.50 per cent molybdenum. All position ½ per cent molybdenum electrodes of ⅛ in. and $\frac{5}{32}$ in. diameter were used throughout. When welded, the pipe samples were approximately 30 in. long.

The inside diameter of each set of pipes was machined at the joint just enough to provide a proper fit for the backing rings. Pipe ends were not bumped up and no recess was made for the backing rings on any of the joints because:

1. Such preparation would materially interfere with the making of reduced section tensile, and face and root bend specimens which would give comparative data on the respective joints.
2. This preparation does not have a definite bearing on the weldability of the joint.

For each of the four groove designs used, one pipe was welded in the fixed horizontal and one in the vertical position. The details of the four groove designs are shown in Fig. 1.

All welds were made in the shop by a production welding operator. The pipe set-up, method of preheating and concurrent heating and welding conditions were typical of shop production conditions. The preheating and concurrent heating gas burner rings and the thermocouples were placed as shown in Fig. 2. A pipe in the vertical position partially welded with gas burner rings covered with asbestos is shown in Fig. 3.

Each pipe sample was heated slowly to approximately 500 F and held there until the joint was completely welded. Each bead was cleaned with a small air operated cleaning tool and hand wire brush before depositing the next bead. Care was taken to insure that each bead was of the proper shape, and in the proper location and that craters were properly filled when the arc was broken for electrode changes. Fig. 4 shows location and shape of beads but in actual pipe welding each bead is com-

---

*Works Laboratory, General Electric Co.
Part 1.

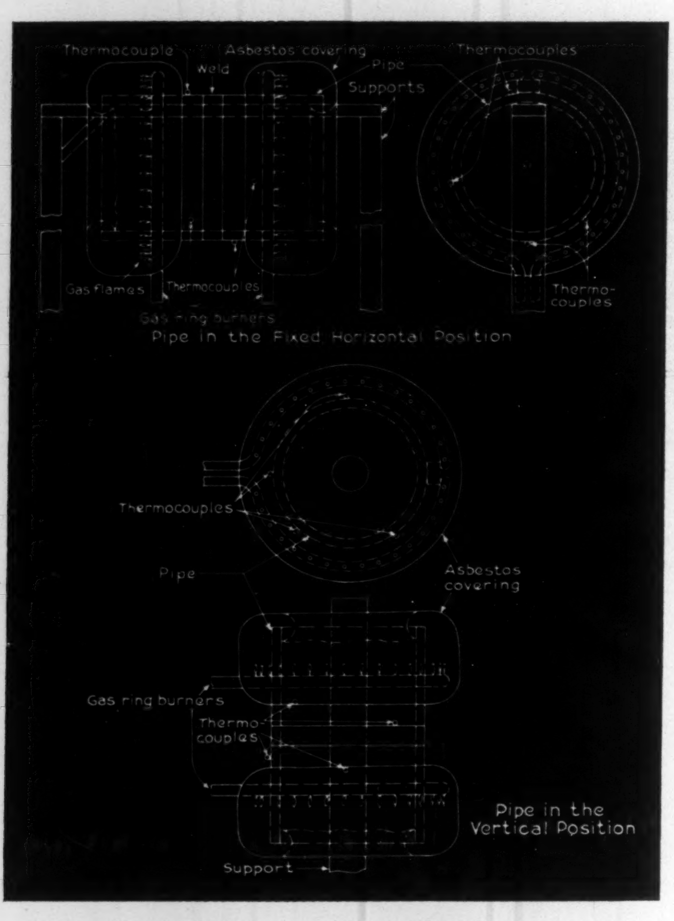

Thermocouple
Asbestos covering
Weld
Pipe
Supports
Thermocouples
Gas flames
Thermocouples
Gas ring burners
Thermo-couples
Pipe in the Fixed Horizontal Position
Thermocouples
Pipe
Asbestos covering
Gas ring burners
Thermo-couples
Pipe in the Vertical Position
Support

Fig. 2—Sketch of pipes set up for welding showing position of gas heating apparatus and thermocouples

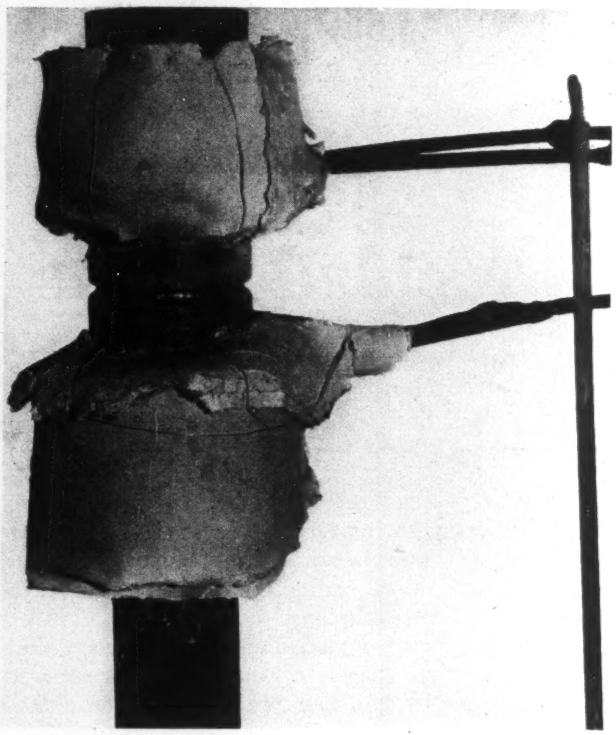


Fig. 3—Pipe set up in vertical position and partly arc welded. View showing asbestos covering and gas supply lines to ring burners used to heat and to maintain desired temperature during welding operation

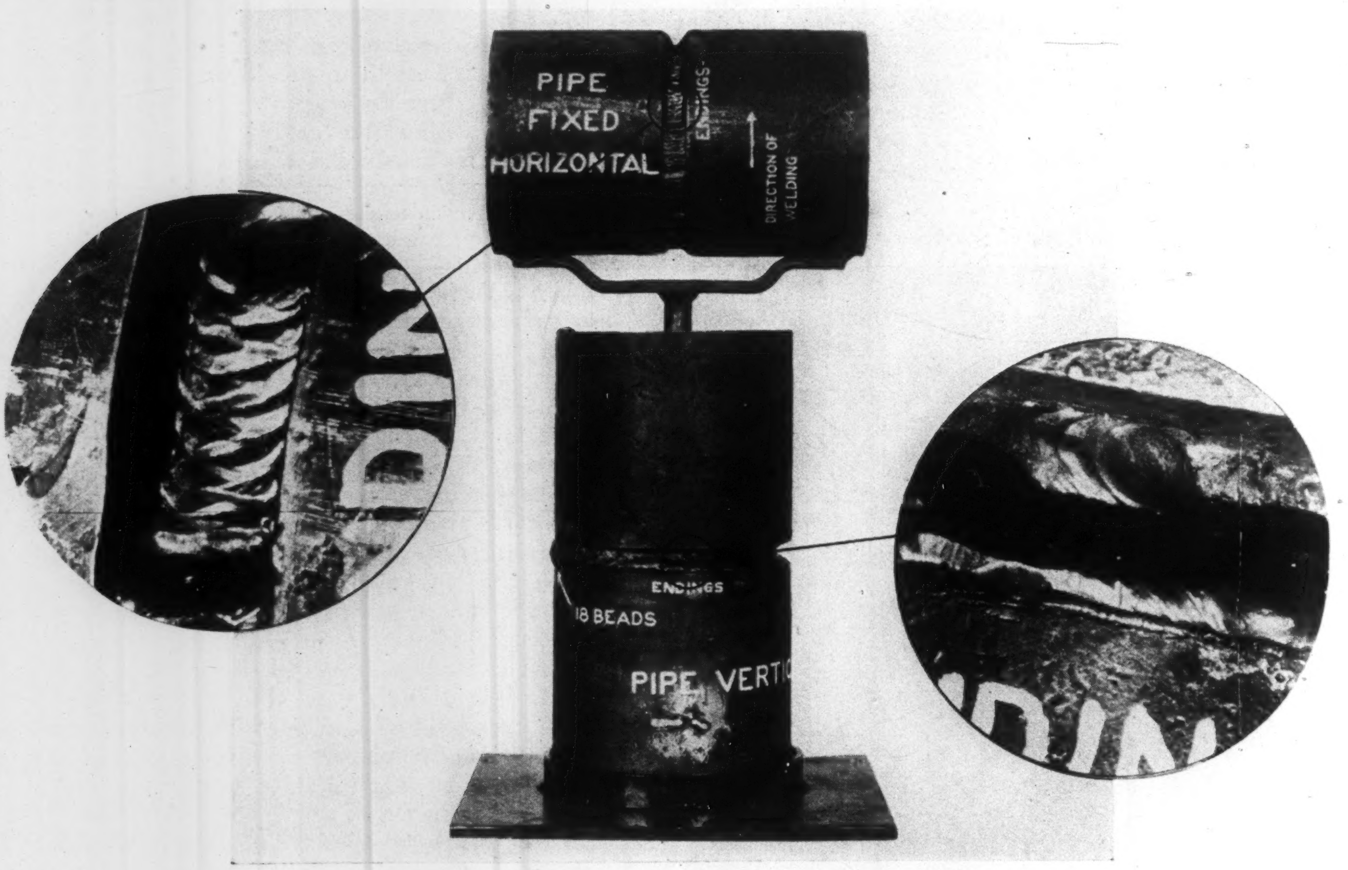

PIPE
FIXED
HORIZONTAL
ENDINGS
DIRECTION OF WELDING
ENDINGS
18 BEADS
PIPE VERTIC

Fig. 4—Two pipe joints welded to show location and shape of each bead

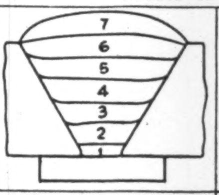
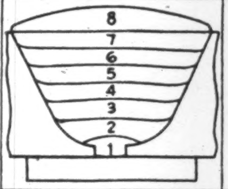
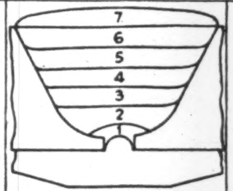
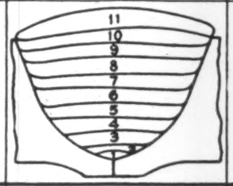
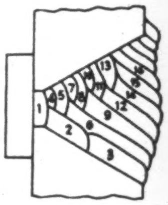
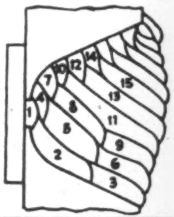
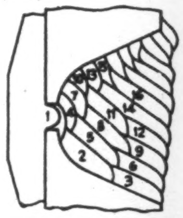
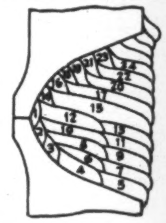
WELDING			PROCEDURE						
Electrode Manipulation	Type, Position, Thickness, and Number of Beads		Size of Electrode	Avg. Current and Arc Voltage 1/8 in Diam Elec	Avg. Current and Arc Voltage 5/32 in Diam Elec	Nature of Cleaning	Direction of Welding	Joint Design	Pipe Joint No.	Fixed Position of Pipe
A whipping action was used in depositing the 1st bead. On all other beads a pool of molten metal was carried and a weaving motion was used.		1. All weave beads 2. Avg. thickness 5/32" - 3. Total No. beads 7	Beads 1 & 2 1/8" diam elec Other beads 5/32" diam elec	80-100 amps 22-26 volts	105-125 amps 23-28 volts	Small air hammer with detachable chisel	Bottom to Top	No. 1	S51-C-12	HORIZONTAL
Same as S51-C-12 Joint Design No. 1		1. All weave beads 2. Avg. thickness 1/8" - 3. Total No. beads 8	Bead 1 1/8" diam elec Other beads 5/32" diam elec	80-100 amps 22-26 volts	105-130 amps 23-28 volts	Small air hammer with detachable chisel	Bottom to Top	No. 2	S51-C-13	
Same as S51-C-12 Joint Design No. 1		1. All weave beads 2. Avg. thickness 5/32" - 3. Total No. beads 7	Beads 1 & 2 1/8" diam elec Other beads 5/32" diam elec	80-100 amps 22-25 volts	105-130 amps 23-28 volts	Small air hammer with detachable chisel	Bottom to Top	No. 3	S51-C-17	
Same as S51-C-12 except that whipping was necessary on the first three beads		1. All weave beads 2. Avg. thickness 1/8" 3. Total No. beads 11	Beads 1,2&3 1/8" diam elec Other beads 5/32" diam elec	80-100 amps 22-25 volts	105-130 amps 23-28 volts	Small air hammer with detachable chisel	Bottom to Top	No. 4	S51-C-18	
A pool of molten metal was carried fairly continuously. In depositing the string beads in the corners, no weaving was possible. In depositing the wider weave beads, a weaving motion was used.	 1 4 5 7 10 13 8 11 16 15 14 12 9 2 6 3	1. Combination of weave and "string beads. 2. Thickness Weave beads 5/32" - String beads 5/32" 3. Total No. beads 20	Beads 1,4&7 1/8" diam elec Other beads 5/32" diam elec	80-100 amps 22-26 volts	105-125 amps 23-28 volts	Small air hammer with detachable chisel	Counter Clockwise	No. 1	S51-C-14	VERTICAL
Same as S51-C-14 Joint Design No. 1	 1 4 7 10 12 14 15 13 8 11 5 9 2 6 3	1. Combination of weave and "string beads 2. Thickness Weave beads 5/32" - String beads 5/32" 3. Total No. beads 24	Bead 1 1/8" diam elec Other beads 5/32" diam elec	80-100 amps 22-25 volts	105-130 amps 23-28 volts	Small air hammer with detachable chisel	Counter Clockwise	No. 2	S51-C-15	
Same as S51-C-14 Joint Design No. 1	 10 13 15 7 4 1 16 14 11 8 12 5 9 2 6 3	1. Combination of weave and "string" beads 2. Thickness Weave beads 5/32" - String beads 5/32" 3. Total No. beads 25	All beads 5/32" diam elec	80-100 amps 22-25 volts	105-130 amps 23-28 volts	Power brush	Counter Clockwise	No. 3	S51-C-16	
Same as S51-C-14 Joint Design No. 1	 23 21 19 18 16 14 24 22 20 17 15 12 10 13 11 8 9 6 7 4 5 1 2 3	1. Combination of weave and "string" beads 2. Thickness Weave beads 5/32" - String beads 5/32" 3. Total No. beads 28	Beads 1,2&3 1/8" diam elec Other beads 5/32" diam elec	80-100 amps 22-25 volts	105-130 amps 23-28 volts	Small air hammer with detachable chisel	Counter Clockwise	No. 4	S51-C-19	

Fig. 5—Summary of welding procedure for each joint welded

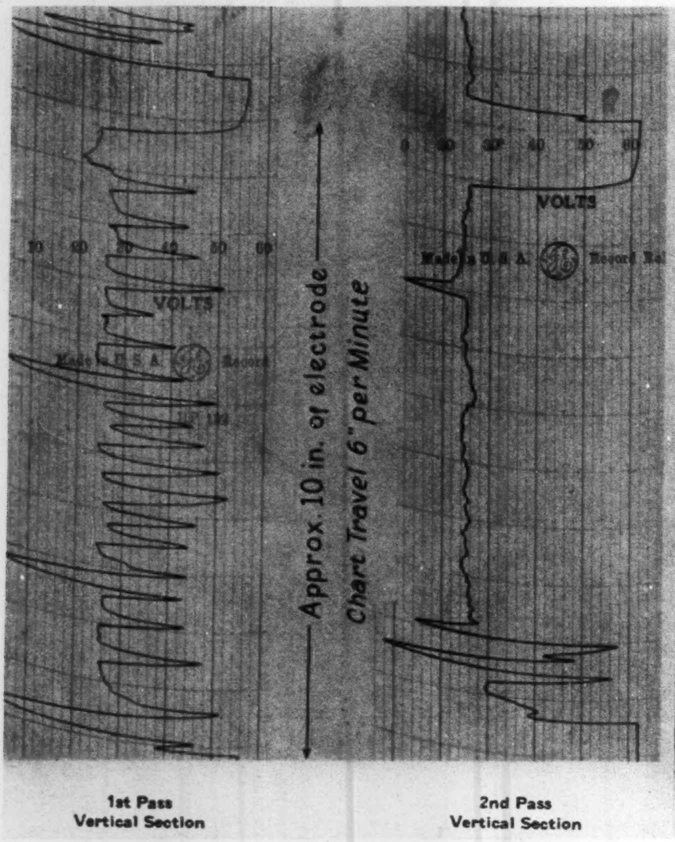


Fig. 6—Typical arc voltage record, pipe position horizontal

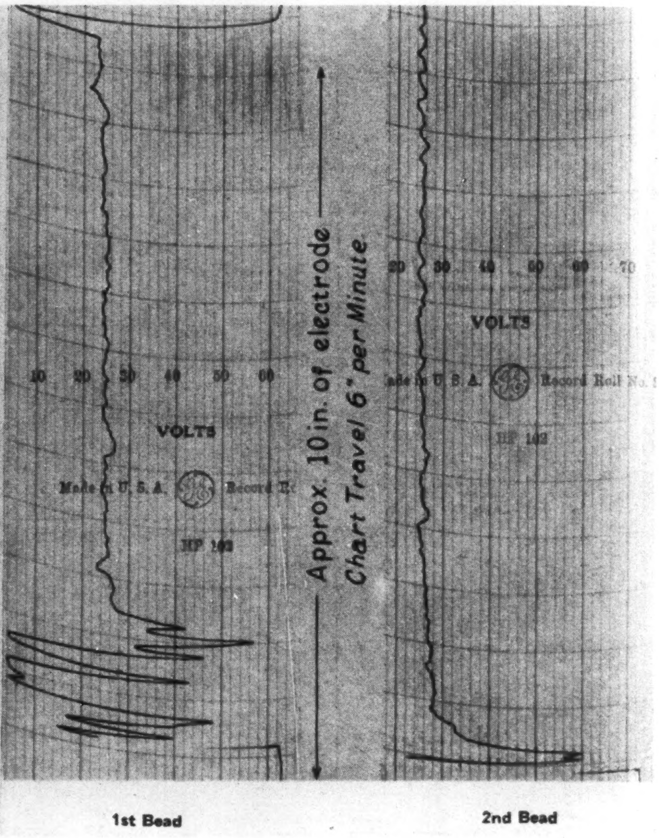


Fig. 7—Typical arc voltage record, pipe position vertical

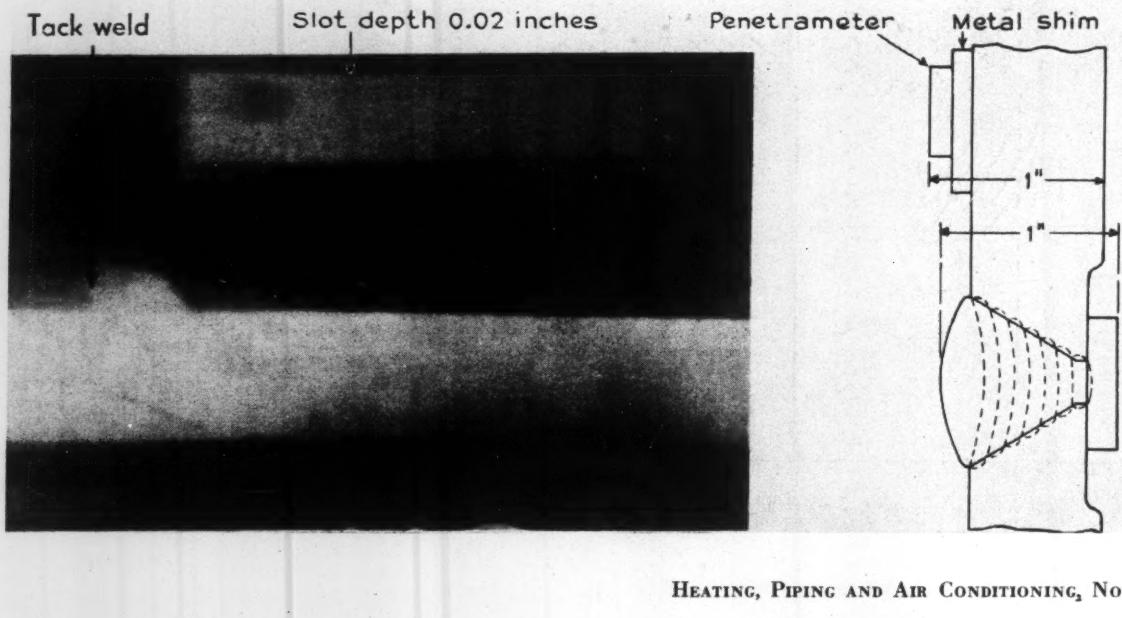
pleted before next bead is started. General chipping of each bead and of craters was not employed. After welding, the pipe sample was covered with asbestos and allowed to cool slowly to room temperature. The welding procedure for each welded joint was as shown in Fig. 5. A recording voltmeter was used throughout the entire welding of samples. Figs. 6 and 7 show typical arc voltage records of first and second beads on fixed horizontal and vertical pipe samples.

Radiographs of three sections of each welded joint were taken. Fig. 8 shows a representative radiograph.

Each sample was locally stress relief annealed by the induction method within a temperature range of 1100 to 1200 F for 1 hr.

From each pipe sample, three reduced section tensile, three face bend and three root bend specimens were saw cut, machined and tested. At least three representative macro-sections of the joint were taken from each pipe sample. These macro-sections were also used for hardness determinations. Fig. 9 shows specimens from pipe welded in the fixed horizontal position after testing. Fig. 10 shows photomacrographs of cross sections of the two welded pipes with hardness values (Rockwell B) and representative photomicrographs.

Fig. 8—Reproduction of radiograph of vertical welded section of fixed horizontal pipe



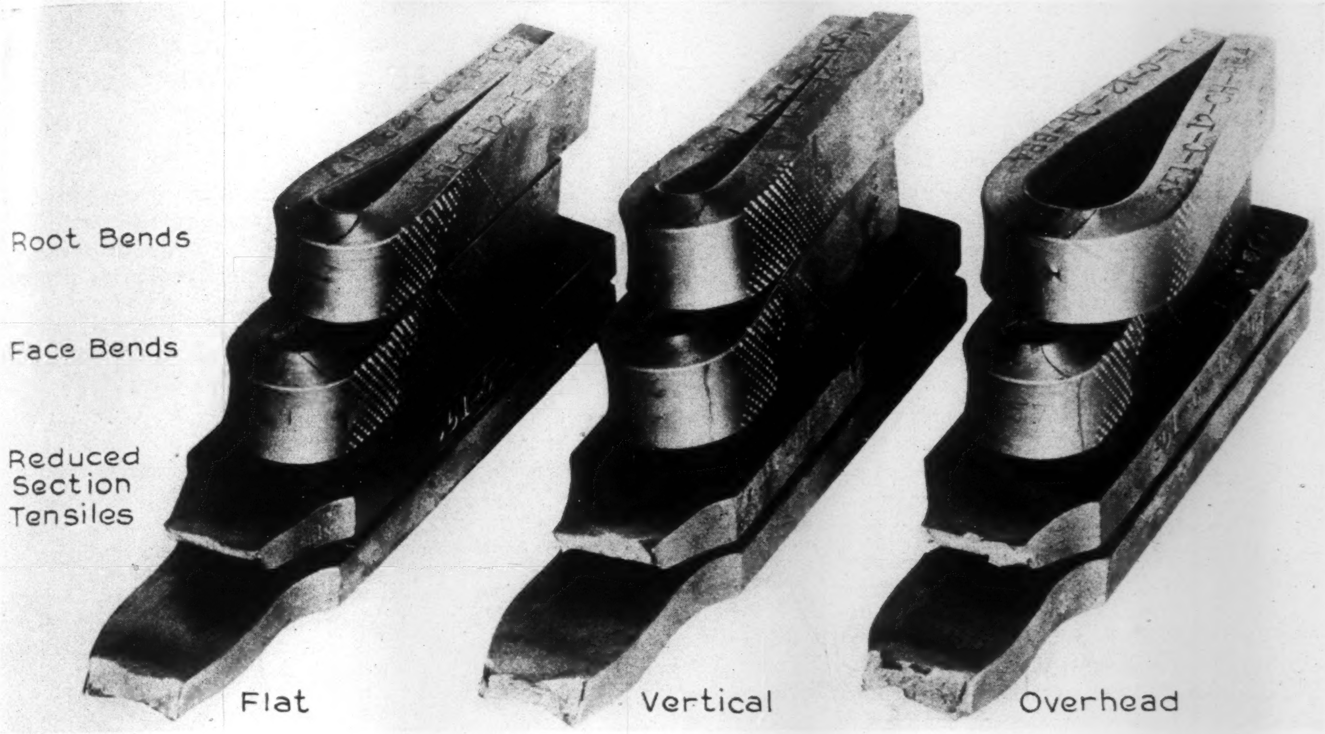


Fig. 9—Typical welded specimens after testing

Examination of radiographs, macro-sections, physical test results and observations during welding show that sound welded joints were made with all four types of groove designs.

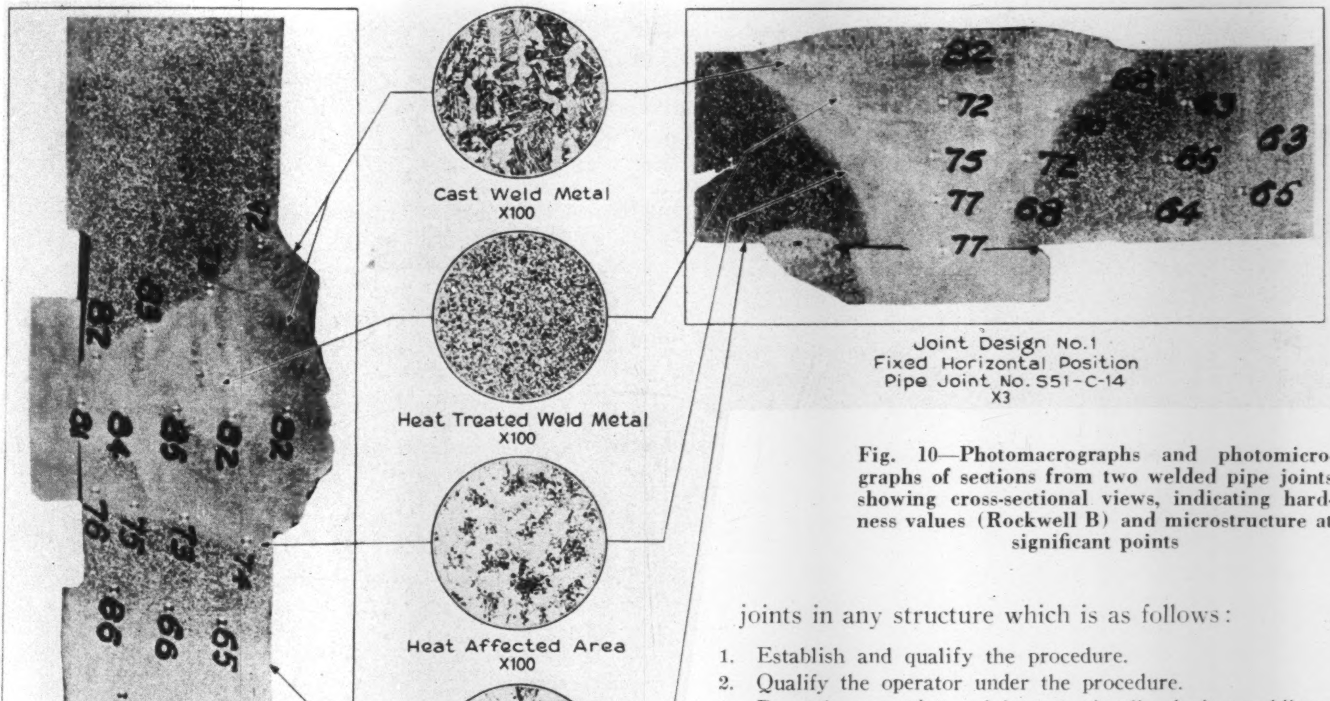
Test results indicate that, with the welding procedure and groove designs used, general chipping of each bead and of craters was not necessary.

In comparing test results with the recording voltmeter records, no defects were found corresponding to the irregularities in the arc voltage records.

These tests substantiate our welding experience with regard to the procedure for insuring satisfactory welded joints in any structure which is as follows:

1. Establish and qualify the procedure.
2. Qualify the operator under the procedure.
3. Properly supervise and inspect visually during welding.

The use of non-destructive tests on completed production welds as a substitute for the above procedure will prove that the above procedure is the most economical and practical when applied before starting production welding.



Joint Design No.1
Vertical Position
Pipe Joint No. S51-C-14
X3

Cast Weld Metal
X100

Heat Treated Weld Metal
X100

Heat Affected Area
X100

Unaffected Parent Metal
X100

Joint Design No.1
Fixed Horizontal Position
Pipe Joint No. S51-C-14
X3

Fig. 10—Photomacrographs and photomicrographs of sections from two welded pipe joints showing cross-sectional views, indicating hardness values (Rockwell B) and microstructure at significant points

[To be concluded]

Movie Makers, Brewers Need Air Conditioning

Comfort air conditioning for Hollywood's super-sound stages was called a necessity by L. Logan Lewis[*] in an address recently before the Society of Motion Picture Engineers. His paper was prepared jointly with Charles M. Wert, air conditioning specialist for the motion picture industry.

The tendency of major producing companies to construct colossal outdoor scenes on indoor stages and the advancement in the art of making backgrounds has brought much of the spectacular "shooting" inside, Mr. Lewis explained. He said through the use of air conditioning on sound stages the term "on location" may be banished from the Hollywood vocabulary.

"Furthermore, familiarity with air conditioning may make it requested among the higher salaried film personnel," he explained. "At the present time Twentieth-Century Fox, Paramount and Metro-Goldwyn-Mayer have a total of 35 air conditioned sound stages."

Mr. Lewis said the increase in size of stages to "football gridiron" area has complicated the need for air conditioning. He stated there are 278 different trades and professions in the motion picture industry and at one time or another a great number of these have representation on the set.

"On large stages some 400 persons have been anticipated and provided for and this number will be exceeded on occasions," he explained. "A number of these occupants are, of course, actors and actresses in heavy costumes and makeup.

"The construction of the modern sound stage plays a part in the necessity for air conditioning, but basically sound stages are being air conditioned because of the need for human comfort; and the results are of economical value.

"The application of air conditioning for the moving picture industry is more allied to human comfort than in any other industrial application."

Two requirements met in air conditioning a sound stage are: 1. The use of air delivered into the stage for actual transportation of heat out of the stage. 2. The furnishing of new air to meet the ventilation requirements of the occupants.

In addition to cooling the studio, the system must circulate, humidify and cleanse the air in summer and supply heating for about nine months of the year. The nights are cool and the stages large; thus heat is required before the actual shooting takes place.

An application of complete air conditioning on the sound stage must meet many demands. It must consist of a flexible system, able at all times to meet the requirement of proper heating, proper ventilation, proper air cleaning, proper cooling, adequate exhaust for the removal of smoke and fog, maintenance of proper conditions of temperature and humidity within the breathing zone and proper distribution of air to enable the meeting of all of the other requirements without discomfort to the occupants.

In addition the systems must be specially built and insulated against all sound, including the swish of the air in the ducts. This is necessary due to the danger of the slightest sound registering on the delicate recording devices.

In concluding, Mr. Lewis emphasized the economic factor as the moving one in the decision to install air conditioning.

"In the moving picture business, as in other businesses, time spent is money spent," he said. "It is obvious that satisfactory results can be produced under comfortable conditions more quickly than under uncomfortable conditions.

"Uncomfortable and unbearable conditions in unconditioned spaces are fought in many ways; doors are opened between shots for flushing the stage with air, shots are delayed, ice cakes and dry ice are brought onto the stage and shots are made at night. All these contrivances lead to a loss of time and money. Damage to costume and makeup from perspiration is self evident and has a dollar value.

"Probably the most conclusive evidence toward the proving of economic value for sound stage air conditioning is the fact that air conditioning systems are still being installed for moving picture companies who have experienced results from previous installations."

Air Conditioning in the Brewery

Control of biological changes which occur during the brewing of beer was named by C. I. Elliott,[†] speaking before the Master Brewers' Association on October 5, as the "number one" job of air conditioning in the brewery. Mr. Elliott said air conditioning was one method through which brewers could keep "these biological changes in the same order and under definite control, thus maintaining uniformity and quality of product."

He listed the starting and fermenting cellars as "musts" for air conditioning. "It is in the starting cellar that the greatest danger of infection exists," he warned. "The wort is at the correct temperature for infection and the period between entering the starting cellar and the time the pitched yeast becomes started in order to build up resistance against infection, is the most vulnerable during brewing.

"The starting cellar should be carefully conditioned, particularly with a view to a sterile and clean condition. It is important to avoid direct currents of air over the starting tub for induced or indirect circulation is better.

"Accurate temperature and humidity are also essential. The natural protective layer of heavy CO_2 on top of the starting tub should not be disturbed. The density of this gas film is such as to cause many small undesirable infectious particles to float off the surface.

"Humidity should be maintained at a point where condensation on the ceiling will be eliminated, since this will collect infectious particles and when falling will drop through the gas film into the wort and thus start infection.

"The conditioning equipment consists of heating and cooling coils, sprays, eliminators, fans and filters with the required controls. One way of circulating the air is delivering the air on the floor under the tubs at a high velocity and taking the return at the ceiling. In this manner the bottoms of the tubs are maintained in a clean condition, corners are dry, foreign yeast spore does not grow in dry places and better control is obtained with the lowest possible air motion over the top of the starting tub."

Of the need for air conditioning in the fermenting cellar having closed or open, wood or steel tanks, he explained:

"Here air conditioning is essential to maintain correct temperature, humidity, air motion and cleanliness. Maintenance of correct temperature in order to allow attemporator coils to be effective, correct humidity to prevent condensation on walls, ceilings or equipment, to dry up floors and equipment after washing, to prevent top yeast from hardening on equipment, to eliminate damp corners or location where wild yeast might grow, supply sufficient ventilation to keep the CO_2 concentration below 3 per cent and for the comfort and safety of workers.

"All outside air should be filtered by the use of efficient filters. The outside air intake should be away from all exhaust or vent pipes. Heating is necessary to assure correct winter temperature and humidities."

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[*] Chief Engineer, Carrier Corp.

[†] Engineer, Carrier Corp.

AMERICAN SOCIETY *OF* HEATING *AND* VENTILATING ENGINEERS

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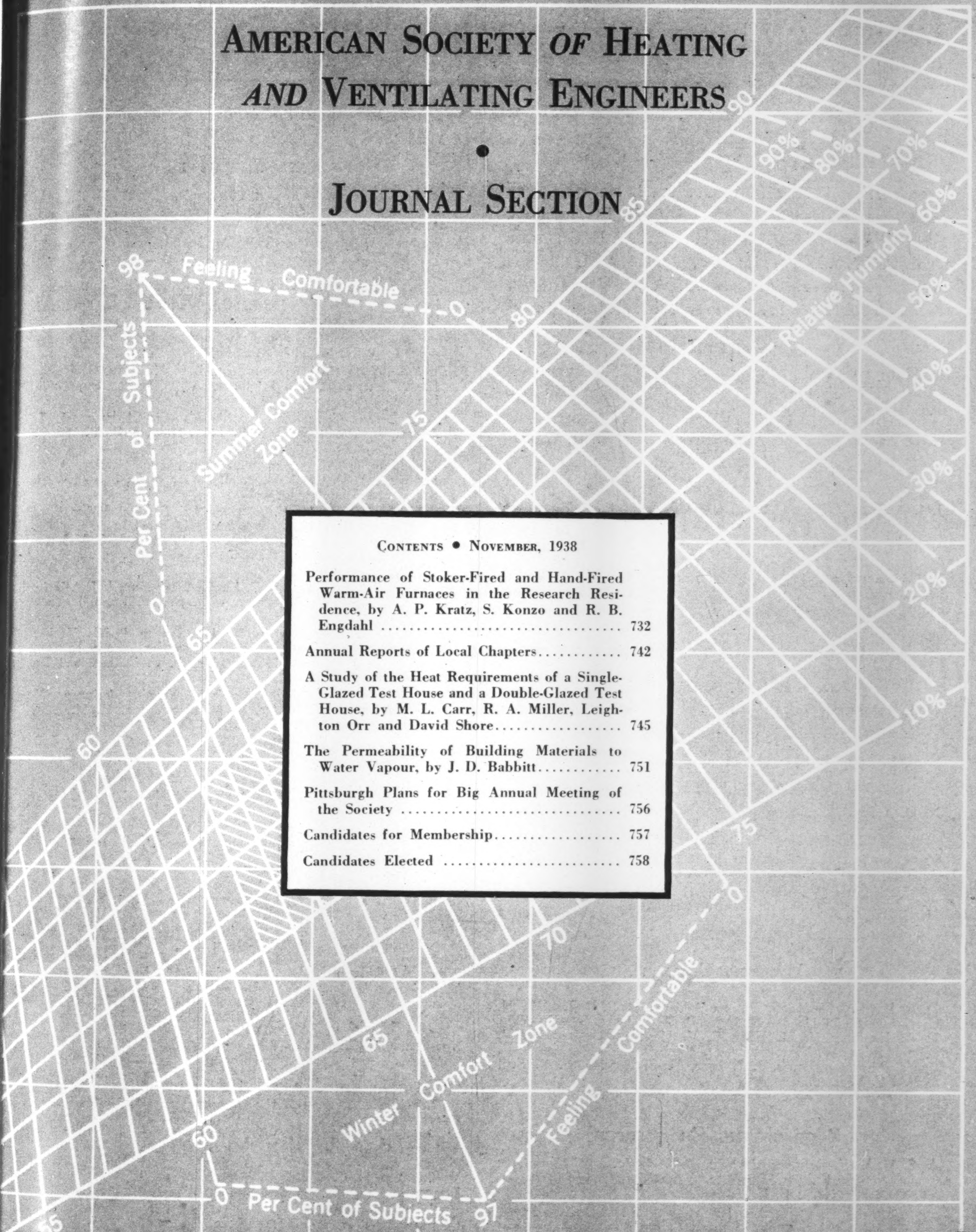

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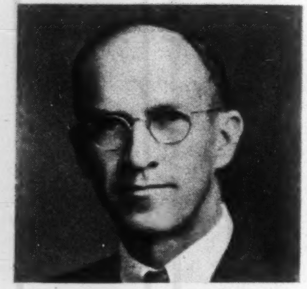
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Performance of Stoker-Fired and Hand-Fired Warm-Air Furnaces in the Research Residence

By A. P. Kratz,* S. Konzo,** and R. B. Engdahl,*** (*MEMBERS*)
Urbana, Ill.

Introduction

Preliminary Statement

THE investigations in forced-air heating in the Research Residence at the University of Illinois during the period from 1932 to 1934 were confined to studies[1] of the characteristics of the forced-air heating system under actual service conditions. For these investigations the fuel used was anthracite. During the period from 1934 to 1937 the performance and operating characteristics were determined[2, 3] of a warm-air furnace equipped with a conversion oil-burning unit and of a warm-air furnace designed specifically for oil firing. During the heating season of 1937-1938, these investigations were extended to include the comparison of the performance and operating characteristics of a thermostatically-controlled hand-fired, warm-air furnace burning a high volatile bituminous coal, with those of the same furnace fired by means of a domestic stoker, of the underfeed type. In order to obtain comparable data no changes were made in the furnace itself, in the plant, in the volume of air circulated, nor in the settings of the thermostat when the stoker was substituted for hand firing. Furthermore, in each case, coal from the same mine and of the size usually recommended for the particular method of firing was used.

Description of the Research Residence and Heating Equipment

The Research Residence in Urbana, Ill., and the heating plant have been completely described in a previous publication.[4] The Residence is a three-story structure of standard frame construction, in which the walls are not insulated, and no weatherstripping is used at the windows nor at the doors. The wall section consists of lap siding, building paper, sheathing, 6-in. studding, wood lath, and plaster with rough sand finish. The windows are double hung, and during all of the tests reported in this paper they remained tightly locked.

The total space heated consisted of three rooms, a sun room, a breakfast nook, and a hallway on the first story; three rooms, a bathroom, and a hallway on the second story; and two rooms, a bathroom, and a hallway on the third story. The total volume of this heated space, from which the basement was excluded, was approximately 17,540 cu ft. The calculated heat losses were approximately 159,000 Btu per hour at an indoor-outdoor temperature difference of 80 F. The Residence is completely furnished, and during the heating season it was occupied by six people.

The heating plant consisted of a warm-air furnace used in connection with a forced-air heating system. Three cold-air returns were used and were connected into a cold-air box above the inlet to a centrifugal type of fan. The furnace was placed at the East end of the basement, and the warm-air registers were served from two main trunk systems.

The furnace used in all of the tests made with both the hand firing and stoker firing was thermostatically controlled, was of the cast-iron circular-radiator type, having a 27-in. firepot and 23-in. grate, and was equipped with a casing 50 in. in diameter. The areas of the heating surfaces and the dimensions of the furnace are presented in Table 1.

In order to install the mechanical stoker, which was inserted through the ash-pit door of the furnace, the grates were removed, and firebrick were arranged to form a flat hearth as shown in Fig. 1. The stoker was of the underfeed type, and coal was delivered from the hopper by means of a rotating screw to the retort, which was located in the middle of the hearth. The hopper had a capacity of 300 lb. Both the rate of fuel input and the rate of air supplied to the tuyeres could be independently regulated. In the case of the tests reported in this paper, the rate of fuel input was maintained at 48.6 lb per hour and the rate of air input was adjusted to provide for satisfactory combustion conditions. In this case the CO_2 content of the flue gas was maintained at approximately 11 per cent when the stoker was in operation, and a slightly hazy atmosphere over the fuel bed was obtained. The rate of air input was not sufficient to blow particles of coal away from the fuel bed.

The rate of burning of the fuel is dependent on the rate of air supply to the fuel bed. It may be observed, however, that in the case of the domestic stoker of the underfeed type the rate of burning of the fuel is not necessarily the same as, and is usually less than, the rate of fuel input to the furnace. If the rate of burning of the fuel in an underfeed stoker were equal to the rate of fuel

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[1] University of Illinois, *Engineering Experiment Station Bulletin* 266, by A. P. Kratz and S. Konzo, 1934.

[2] Performance of Oil-Fired, Warm-Air Furnaces in the Research Residence, by A. P. Kratz and S. Konzo. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1936, p. 693-704.)

[3] Study of Methods of Control and Types of Registers as Affecting Temperature Variations in the Research Residence, by A. P. Kratz and S. Konzo. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1937, p. 745-757.)

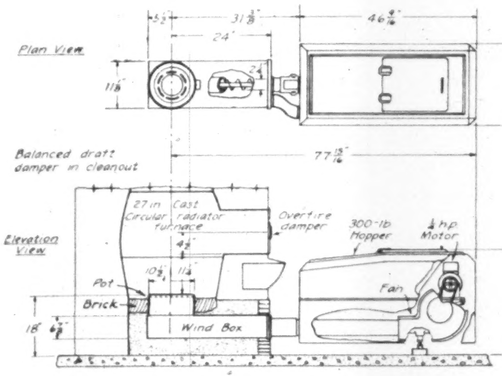
[4] Loc. cit. Note 1.

For presentation at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1938.

Table 1—Dimensions and Areas of Furnace Used

Grate diameter	23 in.
Grate area	2.88 sq ft
Firepot diameter	27 in.
Heating surface	
ash-pit	1.36 sq ft
firepot	8.10 sq ft
dome	18.87 sq ft
radiator	31.93 sq ft
Total	60.26 sq ft
Ratio of heating surface to grate area	20.9
Casing diameter	50 in.
Free area through casing	4.97 sq ft
Combustion space[a]	8.8 cu ft
Free area through over-fire damper	4.9 sq in.

[a] Combustion space is defined in this case as the space above the hearth level, including the dome, but not the feed neck.



5½"
31⅜"
46 9/16"
24"
Plan View
11⅛"
2¼"
22"
Balanced draft damper in cleanout
77 15/16"
27 in Circular furnace
Cast radiator
Over-fire damper
300-lb Hopper
¼ hp Motor
Elevation View
4½"
10½"
11¼"
Pot
Brick
Fan
18"
6½"
Wind Box
29½"

Fig. 1—Diagram of conversion stoker installation in furnace. Research Residence, Season 1937-38, Series 3-37 and 4-37

input, as it is on the chain grate stoker, then no excess fuel would be available in the firepot to maintain combustion during the off-periods. It is probable that with intermittent operation of the stoker, with a given rate of fuel input and depth of fuel bed, some leeway exists in the adjustment of the air volume which will result in satisfactory combustion conditions, and which will at the same time not materially change the level of the fuel bed. Hence, in the case of intermittent operation of the stoker in which some air is inevitably supplied to the fuel bed during the off-period, the rate of burning of the fuel should be such that:

1. Satisfactory combustion conditions are maintained.
2. Sufficient coal remains in the fuel bed when the stoker operation is stopped to allow for combustion during the off-periods of the stoker.
3. Sufficient heat is generated to handle the maximum demands of the house, and
4. The particles of coal are not blown away from the fuel bed.

The fuel burned in these tests was a high-volatile bituminous coal obtained from Saline County, Ill. The characteristics of the fuel are shown in Table 2. In general the fuel used proved satisfactory for both hand firing and stoker firing.

The centrifugal fan in the forced-air system delivered approximately 1675 cfm of air which was recirculated through the house. Six replaceable air filters of the viscous-coated fiber type, each 16 in. x 25 in., were installed on the inlet side of the fan.

The control of the heating plant was accomplished by means of a room thermostat operating to start and stop the stoker motor, or to open and close the ash-pit damper, and to start and stop the circulating fan as shown in Fig. 2. This room thermostat was used in conjunction with two bonnet thermostats which served as high- and low-limit controls for the temperature of the air in the furnace bonnet. The settings for the thermostats are shown in Table 3.

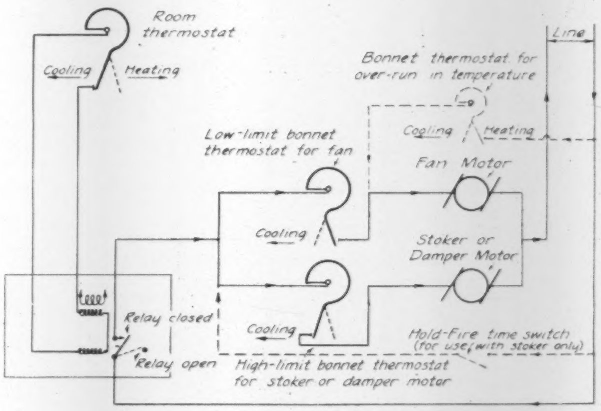
This method of control[5] was highly satisfactory for previous installations in which anthracite and oil fuels were burned, and also proved to be equally satisfactory for both hand firing and stoker firing of bituminous coals. As indicated by the broken lines in the upper right hand portion of Fig. 2, a contact was provided in the bonnet thermostat which closed the fan circuit when the bonnet temperatures exceeded, or over-ran, a temperature of 195 F. This same basic control system was used for the stoker-fired plant and, in addition, an auxiliary switch, designated as the hold-fire time switch, was included. This time-switch is indicated by the broken line in the lower part of Fig. 2. If the bonnet temperature were less than 125 F and if the stoker remained inoperative for 30 min, the time switch closed and the stoker was caused to operate for a two-minute period. This period of operation was sufficient to maintain the fire in the fuel bed, and, except during very mild days when there existed no demand for heat, did not cause any unsatisfactory overheating in the rooms in the house.

The room thermostat, which was of the heat-anticipating type, was located on an inside wall of the dining room at a height 60 in. from the floor, and was adjusted to maintain an average air temperature of approximately 72 F at the 60-in. level in all of the rooms of the Residence.

Method of Conducting Tests

The fan and the damper motor, or stoker, were automatically controlled to maintain an average temperature of 72 F in the rooms of the house both day and night.

Observations of weather, indoor room air temperatures, room relative humidities, and other incidental data were made daily at 7:00 a. m., 11:00 a. m., 4:00 p. m.,



Room thermostat
Cooling
Heating
Line
Bonnet thermostat for over-run in temperature
Cooling
Heating
Low-limit bonnet thermostat for fan
Fan Motor
Cooling
Stoker or Damper Motor
Relay closed
Cooling
Hold-Fire time switch (for use with stoker only)
Relay open
High-limit bonnet thermostat for stoker or damper motor

Fig. 2—Simplified wiring diagram for control of fan motor and stoker motor (or damper motor). Research Residence installation, Season 1937-38, Series 1-37, 3-37 and 4-37

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[5] Control Type IV described in the paper, Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard, (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 37-54.)

Table 2—Characteristics of Coals

General Designation		
Size	Stove (2 in. x 1½ in.)	Stoker (1 in. x 10 mesh)
State	Illinois	Illinois
Vein	No. 5	No. 5
County	Saline	Saline
Type	Non-caking	Non-caking
Treatment	Washed, oil-treated	Washed, oil-treated
Proximate Analysis (As received) per cent:		
Moisture	3.50	8.50
Volatile Matter	38.22	30.96
Fixed Carbon	51.61	53.15
Ash	6.67	7.39
Ultimate Analysis (Dry basis) per cent:		
Carbon	76.30	74.95
Hydrogen	4.90	4.73
Oxygen	8.28	8.94
Nitrogen	1.56	1.47
Sulphur	2.07	1.84
Ash	6.89	8.07
Calorific Value, Btu per pound:		
Coal As Received	13041	12107
Dry Coal	13514	13232
Unit Coal	14670	14550
Air required, pound per pound dry coal	10.3	10.0
Friability Index, D_w, per cent[a]	39.8	39.8
Ash Characteristics, deg F.[b]		
Initial Deformation	1925	1925
Softening Temperature	2000	2000
Fluid Temperature	2193	2193

[a] University of Illinois Engineering Experiment Station Bulletin No. 218.
[b] Average values from Illinois State Geological Survey for Saline County coal.

Table 3—Approximate Thermostat Settings, deg F

	Hand Fired	Stoker Fired
Room thermostat setting	72	72
Lower-limit bonnet thermostat for fan motor		
Make circuit[a]	125	125
Break circuit[a]	110	110
High limit bonnet thermostat for stoker or damper motor		
Make circuit[a]	135	135
Break circuit[a]	150	150
Bonnet thermostat for over-run in temperature		
Make circuit[a]	195	195
Break circuit	190	190
Hold-fire time-switch	...	2 min operation every 30 min

[a] These temperatures are not settings. They correspond to the actual air temperatures in the bonnet outlet as obtained from the temperature recorder in the North Trunk.

and 10:00 p. m. Complete data were obtained for each 24-hour test period on the fuel consumption, weight of ash and clinkers removed, the total integrated time of operation of the fan and of the stoker, the total electrical energy consumption of the fan and of the burner motors, and the total number of on-periods of both the circulating fan and stoker. Daily observations were made of the volume of air circulated, and the filters were cleaned with sufficient frequency to maintain the air volume constant. In addition, continuous records of temperatures and CO_2, such as are shown in the typical records in Figs. 3, 4, and 5, were obtained for each 24-hour period. Continuous records of the index of smoke density were also obtained by means of the apparatus shown in Fig. 6. Observations were made, as will be explained in detail later, on the time consumed in attending the furnace.

For each series of tests, data were obtained for a wide range of outdoor weather conditions, and significant values for each 24-hour period were plotted against the difference in temperature existing between the indoors and outdoors for the same period, as shown in Figs. 7 to 11 inclusive. The plotted points deviate to some extent from the curves representing the average of the observed data, and these deviations can be partly attributed to the wind and sun effects which cannot be represented on curves based on temperature difference alone.

In the first series of tests, which has been designated as Series 1-37, the furnace was fired by hand at 11 a. m., 4 p. m., 10 p. m., and 7 a. m. daily. The fuel bed was poked, grates shaken, and ashes and clinkers removed once each day at 11 a. m. except in extremely mild weather when the fuel bed was disturbed only once every two or three days. Previous to each firing of the fuel the live coals and coke in the fuel bed were shoved back toward the rear of the firepot and the fresh charge of coal was fired in the front. The over-fire damper in the firing door was left wide open at all times. Both the ash-pit damper and the check damper in the smoke pipe were operated by means of the damper motor. With the fuel used, the rate of combustion was found to be extremely responsive to increases in draft. The method of firing employed and the attention given to the furnace may be considered as simulating normally good firing practice.

In the second series of tests, which has been designated as Series 3-37, the furnace was fired by means of the mechanical stoker. At 11 a.m. each day the clinkers were removed, the fuel bed levelled, and the hopper filled with coal. During extremely mild weather no attention was given the fuel bed or hopper, except as required every two or three days. The over-fire damper in the firing door was left open at all times. A balanced check damper was installed in the clean out of the chimney and was adjusted to maintain a draft in the smoke pipe of approximately 0.05 in. of water.

It was observed in the second series of tests that considerable burning accompanied by low CO_2 content of the flue gas occurred during the off-periods of the stoker. Since the burning that occurred during these relatively long periods was undoubtedly augmented by the air entering through the over-fire damper, the third series of tests, designated as Series 4-37, was run with the over-fire damper closed. In all other respects the method of operation was identical with that used in Series 3-37.

Operation of Circulating Fan

For these tests, which were made to study the comparative performance of the hand-fired and stoker-fired furnace plants under actual service conditions, both the circulating fan and coal stoker were operated intermittently by means of the automatic control system, so as to maintain uniform air temperatures in the house at all times. In determining the performance characteristics for each method of operation, data were obtained on the

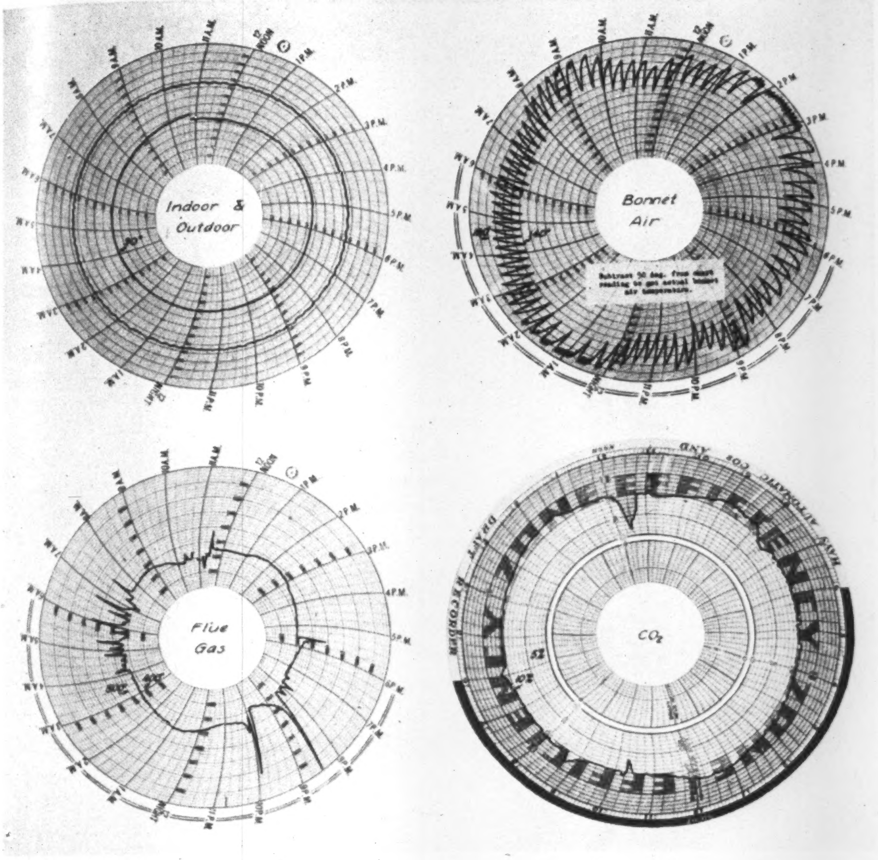

Fig. 3—Temperature and CO_2 records for hand-fired plant (over-fire damper open). Season 1937-38, Series 1-37, Test No. 1256. Forced-air heating system, cast, circular radiator, 27-in. furnace. Bituminous coal, 2 in. x 3 in. stove size

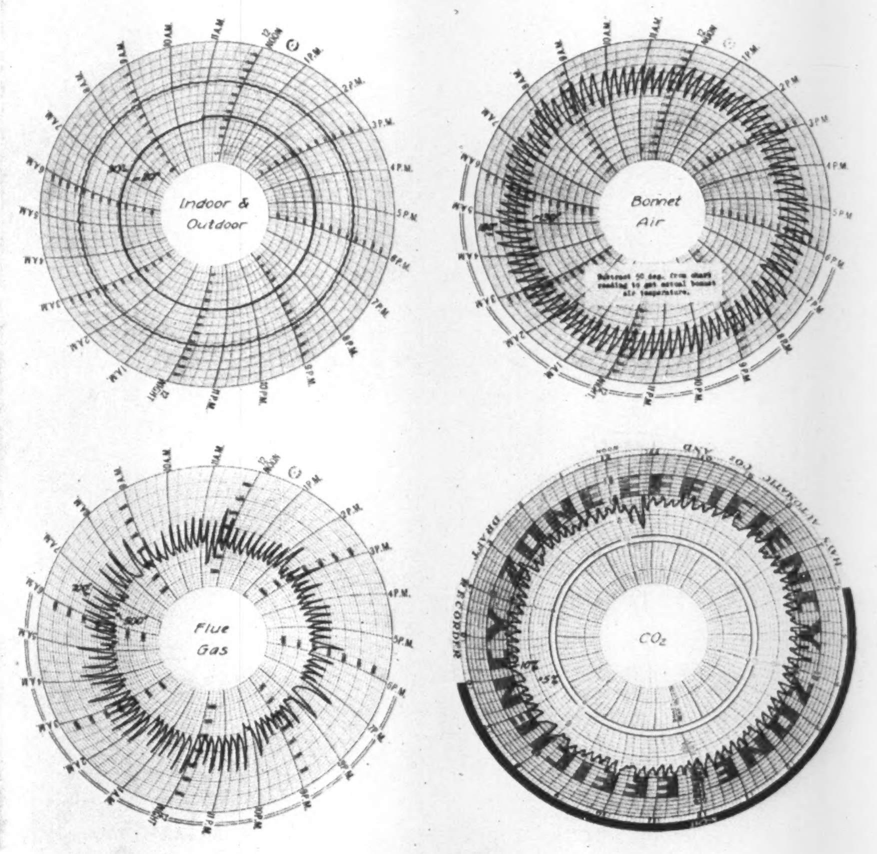

Fig. 4—Temperature and CO_2 records for stoker-fired plant (over-fire damper open). Season of 1937-38, Series 3-37, Test No. 1265. Forced-air heating system, cast, circular radiator, 27-in. furnace. Bituminous coal, 1-in. stoker size

operation of the fan, the consumption of fuel, the operation of the stoker, the time consumed in attending the furnace, and the production of smoke. A comparison of the performance of the heating plant for an average heating day is of interest. For Urbana, Ill., the average outdoor temperature during the heating season is 38 F. Hence for an indoor air temperature of 72 F the value of the average temperature difference between indoors and outdoors is 34 F. The comparisons based on this temperature difference may be regarded as indicative of the results to be secured from the entire heating season.

As shown in Fig. 7, the total number of times the fan was operated, the total hours of fan operation, and the energy consumption of the fan motor per day were each smaller for the hand-fired plant than for the stoker-fired plant. It may be observed that for a temperature difference between indoors and outdoors of 34 F, the energy consumption of the fan was 2.1 and 2.8 kwhr for the hand-fired and stoker-fired plants respectively. This difference can be accounted for by the fact that with the hand-fired plant the mean of the actual bonnet air temperatures was higher than that for the stoker-fired plant, as shown in Fig. 8, and hence the demand of the room thermostat was satisfied by the delivery of warmer air for shorter periods of time.

It should be observed that the actual setting of the bonnet thermostat was the same for all tests, but on account of the larger amount of off-period burning that occurred in the hand-fired plant when the ash-pit damper was closed, the whole range of bonnet air temperatures, and hence the mean bonnet air temperature, that was maintained was higher than that for the stoker-fired plant. The lower limit of the temperature range was high in the case of the hand-fired plant because there was sufficient burning during the off-period, so that, sometimes, when the room thermostat demanded heat, the fan operated. However, the bonnet temperature did not decrease enough to actuate the bonnet thermostat in the damper motor line and thus open the ash-pit damper. When the bonnet air temperature became excessively high, the over-run contact shown in Fig. 2 closed; the fan operated irrespective of whether or not any demand for heat existed, and the bonnet air tempera-

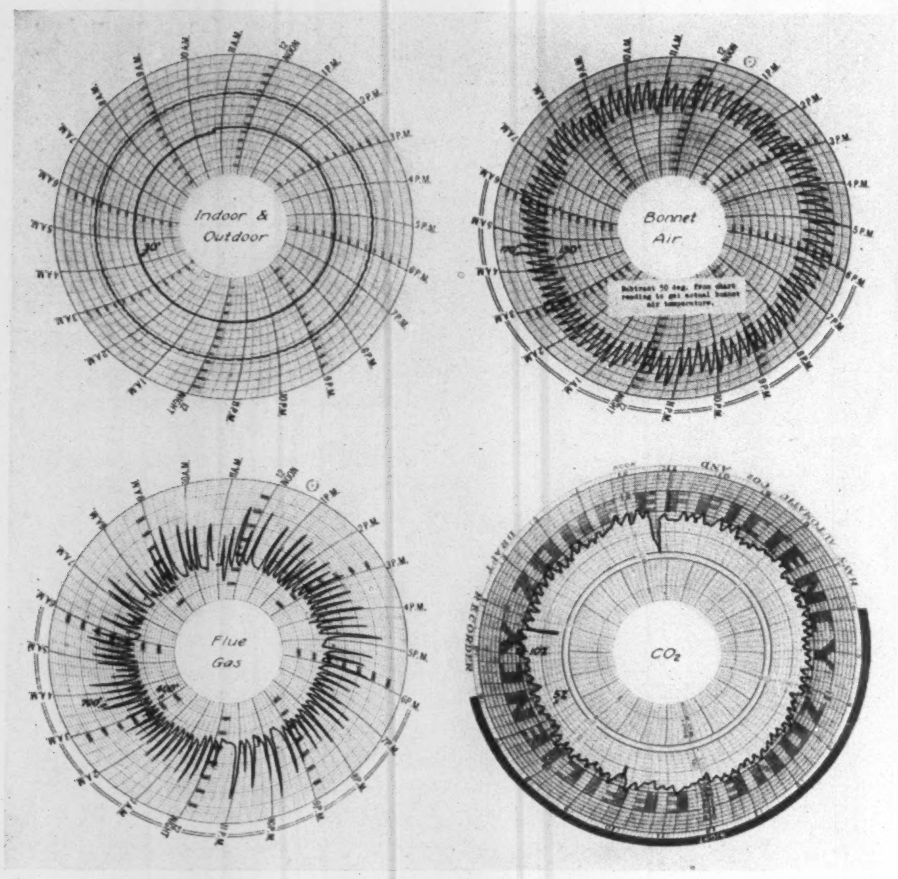

Fig. 5—Temperature and CO_2 records for stoker-fired plant (over-fire damper closed). Season of 1937-38, Series 4-37, Test No. 1311. Forced-air heating system, cast, circular radiator, 27-in. furnace. Bituminous coal, 1-in. stoker size

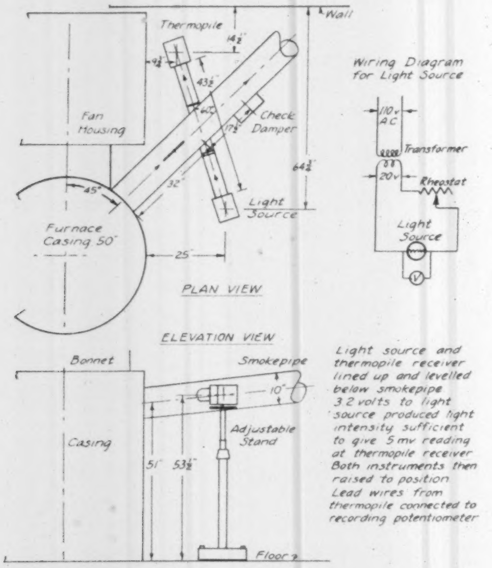
Wall
Thermopile
14½"
9½"
43½"
60°
17½"
Check Damper
Fan Housing
64½"
32"
45°
Light Source
Furnace Casing 50"
25"
PLAN VIEW
Wiring Diagram for Light Source
110 v AC
Transformer
20 v
Rheostat
Light Source
V
ELEVATION VIEW
Bonnet
Smokepipe
10"
Adjustable Stand
Casing
51"
53½"
Floor
Light source and thermopile receiver lined up and levelled below smokepipe 3.2 volts to light source produced light intensity sufficient to give 5 mv reading at thermopile receiver Both instruments then raised to position Lead wires from thermopile connected to recording potentiometer

Fig. 6—Details of installation of smoke density recording instrument in Research Residence, 1937-38

tures were reduced below the upper limit temperature of 195 F. The bonnet air temperatures accompanying these periods of minor cycle operation are shown in the chart for bonnet air temperatures in Fig. 3. The frequency of these minor cycles in total number of operations per day is shown in the upper part of Fig. 7. On account of the slightly greater gravity action during the off-periods of the fan, resulting from the higher bonnet air temperatures, some amount of temperature over-run occurred in the second and third story rooms, even in moderate weather. The temperatures were, however, not unsatisfactory from the standpoint of comfort.

In the case of the stoker-fired plant the combustion rate was greatly diminished when the stoker ceased operation; the bonnet air temperatures were maintained within a lower range, as shown in Fig. 8; and no minor cycles of fan operation resulted. Hence, for moderate or severe weather conditions no overheating occurred in any part of the house. In extremely mild weather the maintenance of the fire by means of the hold-fire switch tended to produce some overheating in the house, particularly in the rooms on the second and third stories.

It is probable that if the combustion during the off-periods had occurred in the same manner for both the hand-fired and stoker-fired plant, the range of bonnet air temperatures would have been the same, and the performance of

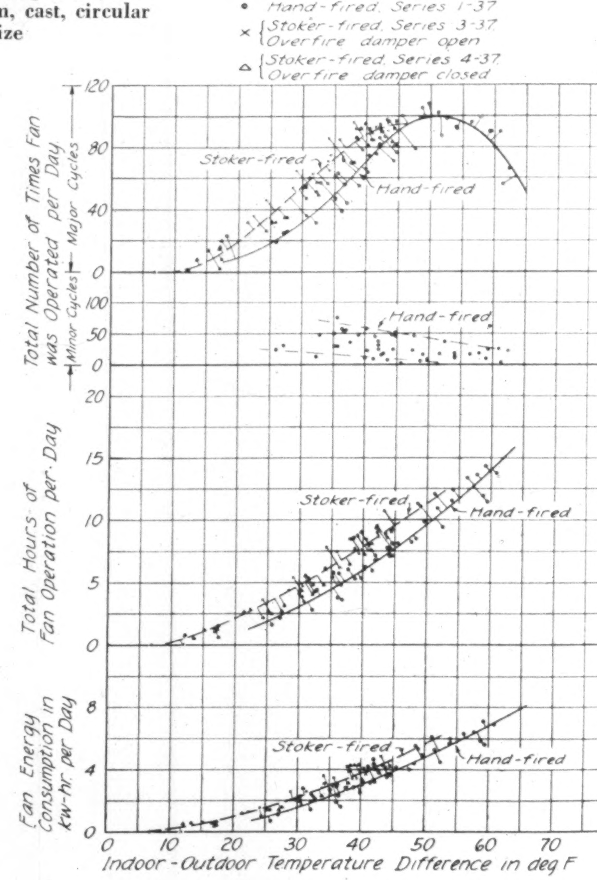

Fig. 7—Operating characteristics of fan. Research Residence tests. Cast, circular radiator, 27-in. furnace. Season 1937-38. Series 1-37, 3-37 and 4-37

the fan would also have been the same in the two cases. In general, the slightly smaller energy consumption of the fan in the hand-fired plant was accomplished by a slight sacrifice in the exactness of the control of the temperatures in the bonnet and in the second and third stories of the Residence. Hence, the reduction in energy consumption of the fan motor cannot be considered as an unqualified advantage.

Consumption of Fuel

Since small deviations occurred in the heating value of the different lots of coal used, the actual weight of coal fired in the stoker-fired plants was corrected to equivalent weight of coal, having a heating value of 13,040 Btu per pound, and the corrected weights were plotted in the curves for fuel consumption shown in Fig. 9. It may be observed from these curves that the weight of coal burned in the hand-fired plant (curve No. 1) was less than that burned in the stoker-fired plant in which the over-fire damper was wide open (curve No. 2), but was greater than that burned in the stoker-fired plant in which the over-fire damper was closed (curve No. 3). The values in Table 4, selected from Fig. 9, for an indoor-outdoor temperature difference of 34 F, show that the weights of coal burned per day were 138 lb for the hand-fired plant, 154 lb for the stoker-fired plant with over-fire damper open, and 128 lb for the stoker-fired plant with over-fire damper closed. Under local conditions a price differential of approximately $1.25 per ton existed in favor of the coal used for the stoker-fired plant.

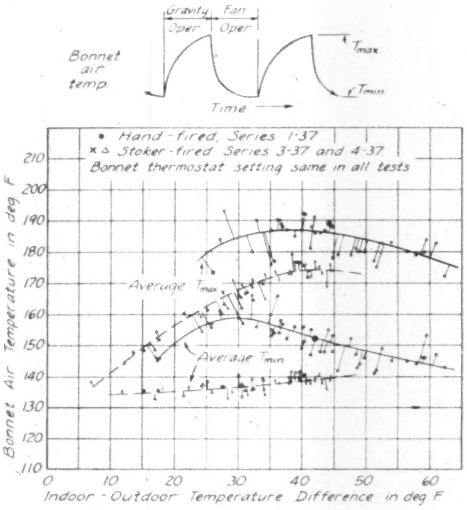

Fig. 8—Operating range of bonnet air temperatures. Research Residence data. Cast, circular radiator, 27-in. furnace. Season 1937-38, Series 1-37, 3-37 and 4-37

In the case of the hand-fired plant the fuel was burned comparatively uniformly, as indicated by the records for CO_2 and flue gas temperatures shown in Fig. 3. On the other hand, in both of the stoker-fired plants the evolution of heat was not as uniform as that for the hand-fired plant, as indicated by the records shown in Figs. 4 and 5. Furthermore, the burning that occurred during the off-periods of stoker operation was accompanied by a lower CO_2 content of the flue gas, particularly when the over-fire damper was open.

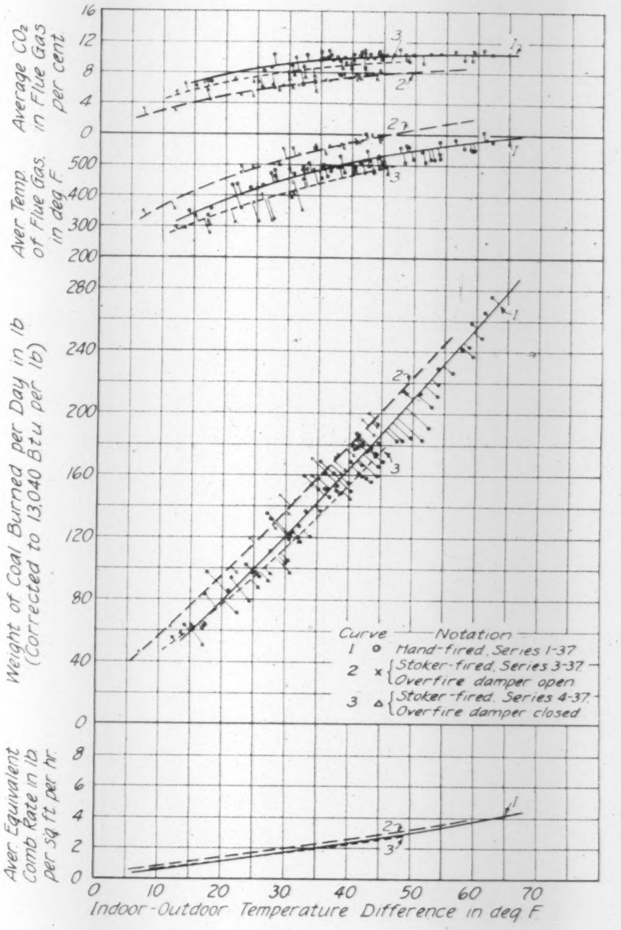

Fig. 9—Performance curves for hand-fired and stoker-fired furnaces. Research Residence tests. Cast, circular radiator, 27-in. furnace. Season 1937-38, Series 1-37, 3-37 and 4-37

Table 4—Combustion Characteristics[a] of Three Methods of Firing

Firing Method	Curve No. in Fig. 9	Weight of Coal Burned, Lb per Day	Average[b] CO_2 in Flue Gas, per Cent	Average[b] Temp. of Flue Gas, Deg F
Hand Firing, Over-fire damper open...	1	138	9.6	480
Stoker Firing, Over-fire damper open...	2	154	6.8	545
Stoker Firing, Over-fire damper closed..	3	128	8.6	440

[a] Values selected from the performance curves in Fig. 9 for an indoor-outdoor temperature difference of 34 F.

[b] Values are average of those obtained during both the on-periods and off-periods of stoker operation.

By proper adjustment of the air supply to the fuel bed during the periods of stoker operation, excellent conditions of combustion could be maintained. During the off-periods of the stoker, however, the fuel remaining in the fuel bed burned with the air supplied through the fuel bed by means of the existing natural draft and with the air supplied over the fuel bed through openings in the feed door. If a large amount of air were supplied from these two sources during the off-periods of the stoker, the amount of coal burned during the off-period would be large.

In the case of the hand-fired plant, the fuel bed contained a large amount of partly coked fuel which con-

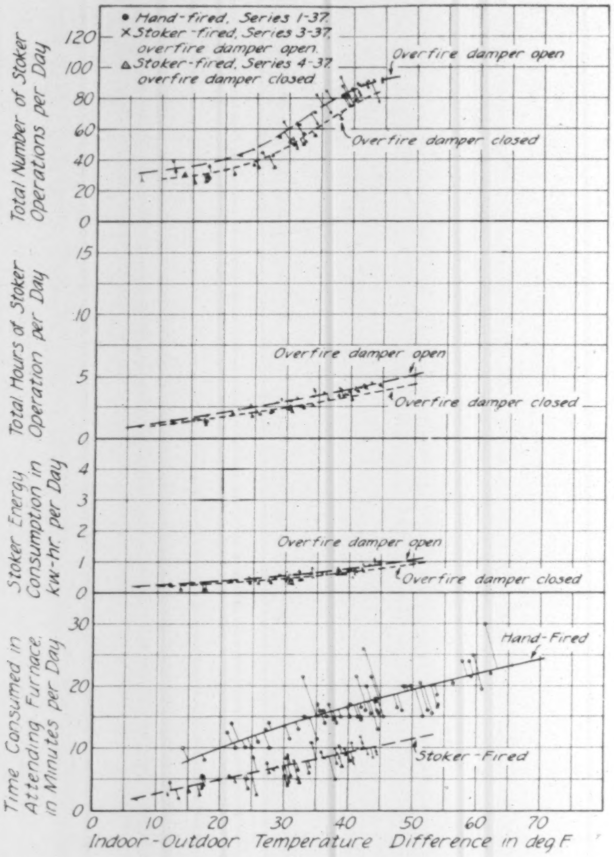

Fig. 10—Operating characteristics of stoker and furnace. Research Residence tests. Cast, circular radiator, 27-in. furnace. Season of 1937-38, Series 1-37, 3-37 and 4-37

tinued to evolve combustible gases during the off-period. These gases required a comparatively large amount of air for combustion, and the amount supplied by the overfire damper was sufficient to promote good combustion of the gases, but did not result in a great amount of combustion of the fuel at the surface of the fuel bed. In the case of the stoker, the fuel remaining at the end of the operating period was well coked, and only a small amount of combustible gas was evolved during the off-period. Hence, with the over-fire damper open a considerably greater amount of air than that required for the combustion of the gases was drawn in, resulting in lower CO_2 content than that shown for the hand-fired plant. At the same time, however, the combustion of the fuel at the surface of the fuel bed was promoted and the evolution of heat from this source, together with the reduced heat transfer caused by the greater mass of air present, resulted in higher flue gas temperatures than those shown for the hand-fired plant.

When the over-fire damper was closed in the stoker-fired plant, the excess air was reduced, resulting in materially higher CO_2 values than those given by the same plant with the over-fire damper open. At the same time the restricted combustion at the surface of the fuel bed resulted in flue gas temperatures much lower than those obtained with the over-fire damper open in the same plant, and somewhat lower than those obtained in the hand-fired plant. The amount of leakage, however, was apparently sufficient, with the small amount of combustible gases evolved, to give more excess air than that obtained in the hand-fired plant, as evidenced by the lower CO_2 obtained in this case than that obtained in the hand-fired plant.

Since the total time of stoker operation per day, as indicated in the second group of curves from the top of Fig. 10, was ordinarily less than five hours, the off-period burning constituted an important part of the entire burning process. The result was that high or low CO_2 contents and temperatures of the flue gas occurring during these off-periods were reflected in corresponding high or low average CO_2 contents and flue gas temperatures as shown in Table 4. Since the flue gas loss is influenced by the combination of CO_2 content and temperature of the flue gas, the higher fuel consumption of the stoker-fired furnace with the over-fire damper open, as compared with that of the hand-fired furnace, is consistent with the lower CO_2 content and higher flue gas temperature shown in Table 4. Also, the lower fuel consumption of the stoker-fired furnace with the over-fire damper closed, as compared with that of the hand-fired plant, is consistent with the slightly lower CO_2 content and the lower temperature of the flue gas shown in this case. This reduction in fuel consumption, however, was accompanied by the production of some smoke and soot, a discussion of which will be presented in a later section.

Operation of the Stoker

The number of stoker operations, the total hours of stoker operation, and the electrical energy consumption of the stoker motor per day are shown in Fig. 10. In each case the values obtained with the over-fire damper open were slightly greater than those obtained with the over-fire damper closed. For an indoor-outdoor temperature difference of 34 F the electrical energy consumption of the stoker motor was 0.6 kwhr per day with the over-fire damper open and 0.5 kwhr per day with it closed. The electrical energy consumption of the stoker motor per ton of coal fired was 7.8 kwhr for both methods of stoker operation. It is probable that with smaller rates of coal input to the furnace the values shown in Fig. 10 would have been larger.

Time Consumed in Attending Furnace

For the purpose of determining the time consumed in attending the furnace, measurements were made by means of a stop watch of the time consumed in shaking grates, removing ashes or clinkers, firing coal into the furnace or hopper, and leveling the fuel bed. The total time consumed per day is shown in the lower group of curves in Fig. 10. These values do not include the time necessary to walk from the first-story living room to the furnace and back. In the case of the Research Residence approximately 1¾ min per trip were required. Hence, if it is desired to account for this time, an addition of 4 x 1.75 or 7.0 min for the hand-fired plant and 1¾ min for the stoker-fired plant may be added to the values shown in Fig. 10.

The total time consumed in attending the furnace on an average day in which the indoor-outdoor temperature difference was 34 F was 15 + 7 = 22 min for the hand-fired plant and 8 + 1¾ = 9¾ min for the stoker-fired plant, or a difference of 12¼ min per day. For a heat-

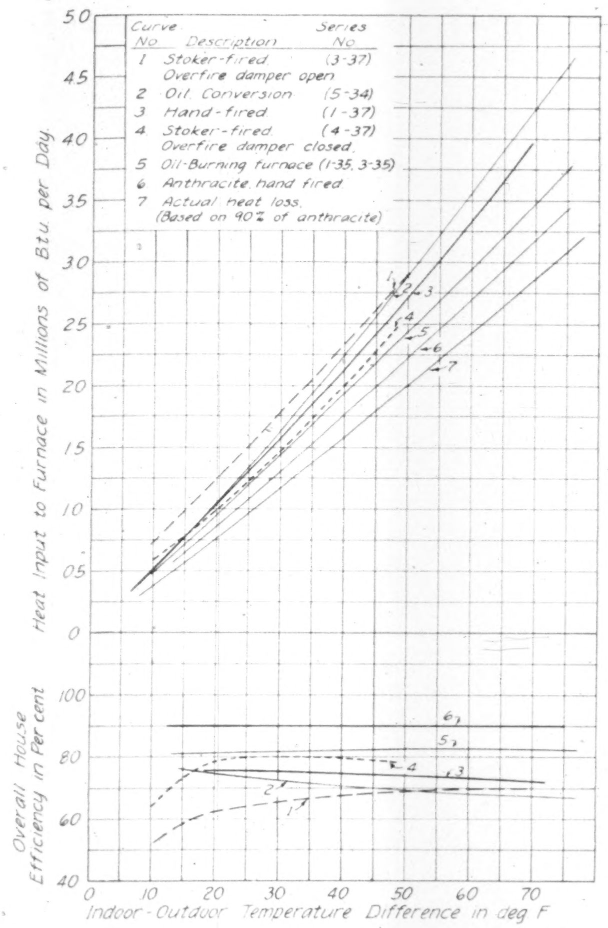

Fig. 11—Heat inputs to Research Residence. Season 1934-38

ing season consisting of 200 days, this difference would amount to

$$\frac{200 \times 12.25}{60}$$

or approximately 41 hours.

Over-All House Efficiency

The significance of over-all house efficiency was discussed in a previous publication,[6] and further data on this subject were obtained on the stoker tests. In order to compare the fuel consumption with oil, anthracite, and bituminous coal, the fuel quantities shown in Fig. 9 were reduced to terms of heat input to the furnace, in millions of Btu per 24 hours, and were plotted as shown by curves Nos. 1, 3, and 4 in Fig. 11. The values for oil, (curves Nos. 2 and 5) and anthracite (curve No. 6) have been reproduced from a previous paper.[7] In all cases the operation of the heating plant was controlled by means of sensitive thermostatic controls.

From analysis of the flue gases when anthracite was used as a fuel, the heat lost in the gases escaping from the chimney, which represented heat furnished by the fuel that was not ultimately available for heating the house, was determined as approximately 10 per cent of the total heat input to the furnace. The actual heat loss from the house (curve No. 7) was then derived from the experimental results for heat input obtained with anthracite as a fuel (curve No. 6). By using these derived values of the actual heat loss from the house in connection with the fuel consumption curves for the hand-fired and stoker-fired plants, the over-all house efficiency, or the ratio of the heat loss from the house to the heat input to the furnace, could be calculated as shown in the lower part of Fig. 11. For an indoor-outdoor temperature difference of 34 F the over-all house efficiencies were 80 per cent with the stoker-fired plant having over-fire damper closed, 75 per cent with the hand-fired plant, and 67 per cent with the stoker-fired plant having over-fire damper open. The value of 75 per cent obtained with the hand-fired plant using bituminous coal is in substantial agreement with results[8] previously obtained in the Research Residence.

The results obtained from these tests were to a great extent dependent upon the method of operation used. The conditions of hand firing which were maintained simulated normal intelligent firing of a thermostatically-controlled hand-fired plant. In the case of the hand-fired plant, it is probable that the fuel consumption would have been larger if lump coal, rather than sized coal adapted to hand firing, had been used; if hand control, rather than automatic thermostatic control, had been used; or if the coal had been thrown on top of the fire rather than to one side.

Smoke Density

The smoke density meter, consisting of a lamp and a thermopile, as shown in Fig. 6, used in connection with a recording potentiometer, provided a means for obtaining continuous records of the index of smoke density in the smoke pipe. The lamp voltage was adjusted, by means of a rheostat, so that when no smoke passed through the smoke pipe a maximum reading of five millivolts was obtained from the thermopile upon which the light beam was received. Smoke passing through the smoke pipe reduced the intensity of the light beam reaching the thermopile. When the light beam was completely intercepted the thermopile gave a reading of zero millivolts.

For the purpose of calibrating the instrument in terms of the more familiar Ringelmann numbers, the electromotive force generated by the thermopile was observed by means of the potentiometer, and at the same time the

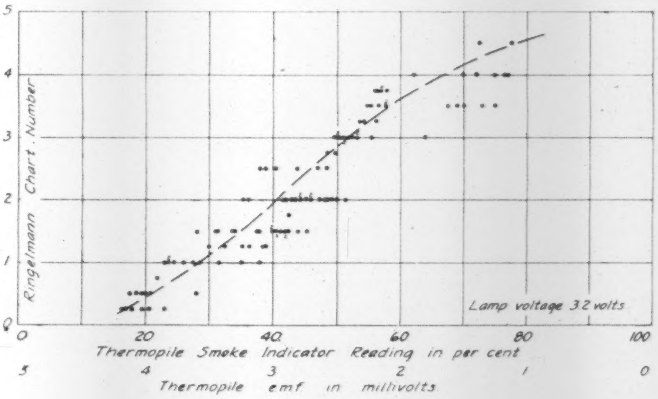

Fig. 12—Calibration of smoke density recording instrument

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[6] Investigation of Warm-Air Furnaces and Heating Systems, Part IV, by A. C. Willard, A. P. Kratz, and V. S. Day, (University of Illinois, *Engineering Experiment Station Bulletin* 189, Chapter V.)

[7] Loc. cit. Note 2.

[8] Loc. cit. Note 6.

smoke emitted from the top of the chimney was compared with Ringelmann charts. The calibration curve for the instrument, with the check draft sealed in order to prevent air leakage between the instrument and the top of the chimney, is shown in Fig. 12. It may be noted that an indicator reading of 2.4 millivolts was obtained with smoke density corresponding to No. 3 Ringelmann. It is common practice for smoke ordinances to limit the production of smoke to six minutes of No. 3 smoke in an hour.

The index of smoke density, plotted against the time in hours after the coal was fired, is shown for the hand-fired furnace in the upper part of Fig. 13. The scale of Ringelmann numbers is given on the right hand side of the graph. It was observed that, when coal was added to the fuel bed at a time when low combustion rates were maintained, the comparatively low temperatures of the fuel bed resulted in correspondingly small rates of evolution of the gases and of formation of smoke. This condition is shown by the curve designated as the *upper range* in Fig. 3, and it may be observed that in this case the recommended limit of No. 3 smoke for a period of six minutes was never exceeded. However, when coal was added to a comparatively hot fuel bed, it was rapidly volatilized and large quantities of smoke were evolved for a period of approximately 15 min after the coal was fired, as shown by the curve designated as the *lower range* in Fig. 13. Under these conditions the smoke given off by the fresh charge of coal exceeded the maximum limit of six minutes per hour of No. 3 Ringelmann density. In general, the amount and duration of the smoke produced by the hand-fired furnace were largely dependent upon the temperature of the fuel bed at and immediately following the time of firing. After the gases had been driven off from the fresh charge of coal, and the fuel had been coked, the density of smoke was ordinarily less than No. 2 Ringelmann.

In both cases with stoker firing, as shown in the lower diagrams in Fig. 13, the density of smoke during the on-periods of the stoker was less than No. 1 Ringelmann and could be considered as practically negligible. During the period of approximately 30 seconds after the stoker was stopped the smoke density increased sharply to attain a maximum value of about No. 3 Ringelmann. It then decreased rapidly during the succeeding two minutes. For a few seconds in each hour a smoke density of No. 3 Ringelmann was obtained. The smoke conditions accompanying the use of the stoker-fired furnace with the over-fire damper open could be considered as entirely satisfactory. However, with the over-fire damper closed, the combustion which occurred during the off-periods of the stoker was accompanied by some evolution of smoke and the formation of soot which coated the heating surface. It is probable that operation of the furnace for an entire season under these conditions would have resulted in some decrease in effectiveness of heat transmission and in some difficulties in maintaining proper draft. Hence, although the closing of the over-fire damper resulted in better fuel economy, this economy was obtained by a sacrifice of clean heating surfaces. Some amount of over-fire air was necessary in order to prevent the accumulation of soot. It is probable that the amount of over-fire air necessary would to a great extent be dependent upon the type of coal and the type of furnace and stoker used.

Incidental Data

With the cooperation of the Illinois State Geological Survey, motion pictures of the fuel bed in the stoker-fired plant were made. Typical photographs showing various stages in the combustion process were made from enlargements of the original film and are shown in Fig. 14. Figs. 14a and 14b show conditions common to both series of tests. Figs. 14c and 14d refer to conditions with the over-fire damper open, and Figs. 14e and 14f to those with the over-fire damper closed. None of these photographs shows any evidence of coke-tree formations. Furthermore, no difficulty was experienced from failure to form clinkers. Fig. 14a represents the fuel bed immediately after the stoker started, and shows active combustion starting in the center. The clarity of details indicates the entire absence of smoke. Fig. 14b was taken after the combustion had progressed for five minutes, and shows active and smokeless combustion occurring over the major part of the fuel bed. Fig. 14c shows that, immediately after the stopping of the stoker and with the over-fire damper open, the combustion in the fuel bed continued rather actively with air furnished through the over-fire damper, and that a slight amount of smoke was formed. Fig. 14d taken five minutes after the stoker was stopped shows that, with the over-fire air damper open, combustion in the fuel bed continued even during the off-period of the stoker. It may also be noted that the production of smoke had ceased by this time.

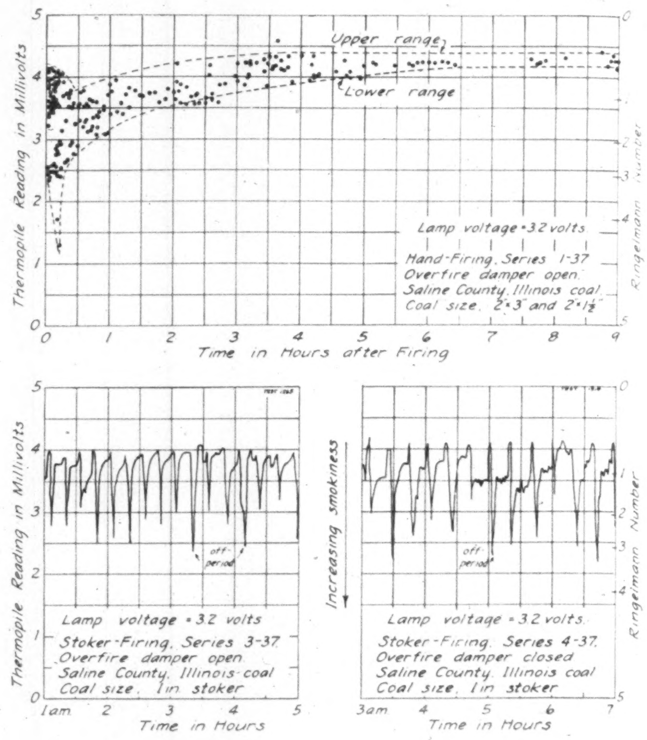
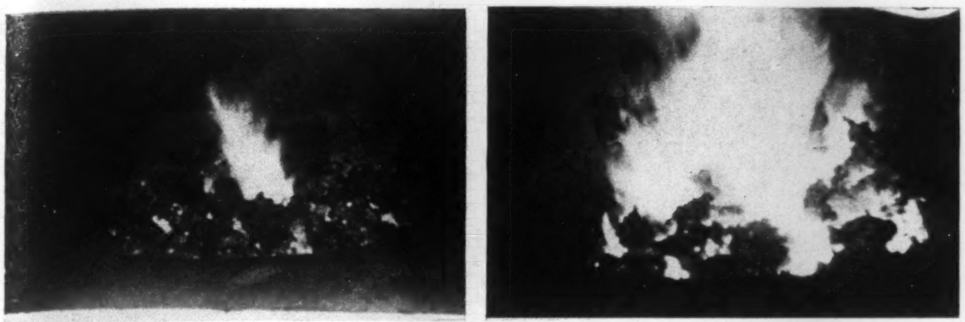


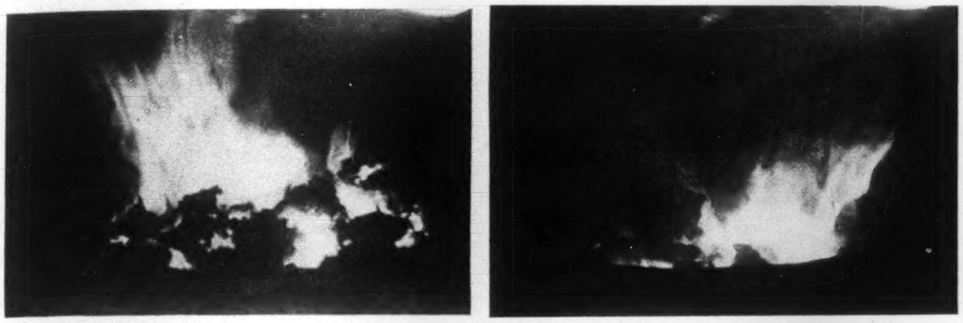
Fig. 13—Smoke density records for hand-fired and stoker-fired furnace. Research Residence data. Season 1937-38, Series 1-37, 3-37 and 4-37

Fig. 14e shows that, immediately after stopping the stoker with the over-fire damper closed, combustion again continued in the fuel bed. However, in this case the reduction in the amount of over-fire air resulted in the formation of a considerable amount of smoke, and the evolution of smoke continued for five minutes after



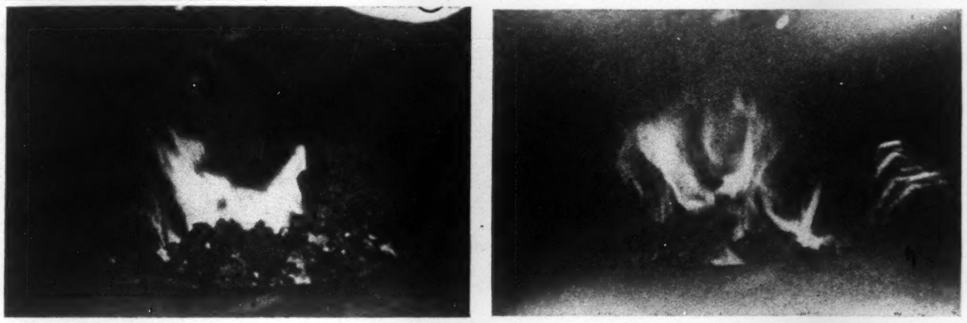
(*a*) Beginning of stoker operation

(*b*) After five minutes of stoker operation



(*c*) Beginning of off-period, over-fire damper open

(*e*) Beginning of off-period, over-fire damper closed



(*d*) Five minutes after stopping stoker, over-fire damper open

(*f*) Five minutes after stopping stoker, over-fire damper closed

Fig. 14—Photographs of stoker fuel bed

the stoker was stopped, as shown by Fig. 14f. Even in this case combustion continued during the off-period of the stoker, but not as actively as it did with the over-fire damper open.

An inspection of the radiator, the smoke pipe, and the chimney clean out indicated that in the case of the stoker-fired plant considerable amounts of fly ash were deposited. The ash per pound of coal fired (theoretical ash) was 7.39 per cent; whereas the actual weight of all the ash and clinkers, together with any unburned combustibles, removed from the fuel bed was 7.00 per cent by weight of the coal fired. A chemical analysis of the ash and clinkers removed from the furnace indicated that only a trace of unburned combustibles (less than 0.05 per cent) remained in the ash. Hence, the difference of 0.39 per cent of the weight of coal fired may be considered as an index of the amount of ash, in the form of fly ash, that was not removed through the feed door. This represents an amount equivalent to 0.39 lb of fly ash per 100 lb of coal fired, or 7.8 lb per ton of coal fired. In the case of the hand-fired plant the difference between the weight of the ash and refuse and the theoretical weight of ash was approximately five per cent. Since no appreciable amount of fly ash was in evidence, this five per cent may be regarded as combustible lost in ash.

These preliminary studies with a stoker-fired furnace have indicated that the results obtained are dependent upon the adjustments made in the operating conditions. Further tests are contemplated, particularly with reference to the effect of varying the rate of coal input and the rate of burning on the over-all plant performance.

Summary and Conclusions

The following summary and conclusions may be considered as applying to the Research Residence and the conditions under which the tests were conducted.

(1) The method of control used, consisting of a room thermostat and a bonnet thermostat which operated both the circulating fan motor and the stoker motor, was found to be highly satisfactory for a forced-air heating plant fired by an underfeed stoker.

(2) With a given indoor-outdoor temperature difference, the weight of coal burned in the hand-fired plant was less than that burned in the stoker-fired plant in which the over-fire damper was wide open, but was greater than that burned in the stoker-fired plant in which the over-fire damper was closed.

(3) The burning that occurred during the off-periods of stoker operation constituted a relatively important part of the entire burning process.

(4) For a coal input rate of 48.6 lb per hour, the electrical energy consumption of the stoker motor was 7.8 kwhr per ton of coal fired.

(5) The total time consumed in attending the furnace on an average day was 22 min for the hand-fired plant and 9¾ min for the stoker-fired plant.

(6) During the period of approximately 30 seconds after the stoker was stopped, the smoke density increased sharply to attain a maximum value of about No. 3 Ringelmann and then decreased rapidly during the succeeding two minutes.

(7) With the over-fire damper closed the combustion which occurred during the off-periods of the stoker was accompanied by some evolution of smoke and by the formation of considerable amounts of soot.

(8) In stoker-fired furnaces it is advisable to provide accessible clean-out doors for the removal of any fly ash which may accumulate in the gas passages.

Acknowledgments

The results presented in this paper were obtained in connection with the investigation of warm-air furnaces and heating systems in the Research Residence,‡ at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. This investigation is a cooperative project sponsored jointly by the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgments are also due to Dr. Louis C. McCabe, and Dr. Harold B. Willman, Associate Geologists of the Illinois State Geological Survey, for their assistance in making motion pictures of the fuel bed. The assistance of the various manufacturers who cooperated by furnishing instruments and equipment is gratefully acknowledged.

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‡The Research Residence in Urbana, Ill. was built, furnished and completely equipped specifically for research work in warm-air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

ANNUAL REPORTS OF LOCAL CHAPTERS

Atlanta

Atlanta Chapter has actively started its 1938-39 season under the following officers and committees, who are in charge:

President—C. L. Templin.
Vice-President—S. W. Boyd.
Secretary—T. T. Tucker.
Treasurer—C. B. Cole.
Board of Governors—C. T. Baker, E. W. Klein.
Finance Committee—C. B. Cole, chairman; Leo Sudderth, Jr., vice-chairman; F. W. Clare.
By-Laws Committee—W. J. McKinney, chairman; M. G. Driscoll, vice-chairman; R. F. Hahn.
Membership Committee—E. W. Klein, chairman; C. E. Brockinton, vice-chairman; M. M. Crout.
Entertainment and Program Committee—L. L. Hamilton, chairman; F. L. Laseter, vice-chairman; R. E. Nolan.
Codes Committee—W. P. Sherman, chairman; S. W. Boyd, vice-chairman; C. T. Baker.
Publicity Committee—G. H. Brodnax, Jr., chairman; L. F. Kent, vice-chairman; R. D. Kelley.

The Chapter numbers 28 on its membership roll and during the past 12 months ten business meetings and two social gatherings were held.

October 12, 1937. D. S. Boyden, Boston, Mass., president of the A. S. H. V. E., gave an illustrated talk on the Various Uses of Different Means for Heating Buildings. Preceding the talk, a dinner was given in honor of President Boyden, who later presented the Chapter with its charter. Guests included faculty members, as well as part of the student body, from Georgia School of Technology.

November 3, 1937. This was a regular business meeting and it was decided to consider papers by local members for presentation at future meetings.

January 11, 1938. W. A. Evans, Aerofin Corp., Atlanta, gave an interesting paper on Unit Heaters and Coolers.

February 8, 1938. Reports from Pres. E. W. Klein and L. F. Kent concerning the Society's Annual Meeting in New York were presented.

March 1, 1938. M. G. Driscoll, Bryant Heater Co., Atlanta, gave an interesting paper on the Properties and Uses of Silica Gel.

April 8, 1938. Albert Buenger, Delco-Frigidaire Conditioning Div., Dayton, O., addressed the Chapter on Reducing the Cost of Air Conditioning. Mr. Buenger was chairman of the 1938 Guide Publication Committee of the A. S. H. V. E.

April 23, 1938. Prof. F. E. Giesecke, second vice-president of the Society, College Station, Tex., gave an instructive paper on the Influence of Building Construction on Air Conditioning.

May 4, 1938. This meeting was taken up with the election and installation of new officers.

June 15, 1938. A social meeting was held at East Lake, jointly with the *A. I. E. E.* and *A. S. M. E.* local groups, and the program included golf and dancing.

July 16, 1938. This was the Chapter's athletic meeting held near a small lake on the outskirts of Atlanta.

Northern Ohio

In the annual report of the Northern Ohio Chapter, Prof. C. A. McKeeman calls attention to the successful 1937-38 season as evidenced by attendance records and interest shown in the topics discussed at the various meetings. The Chapter looks forward to another active season under the leadership of the following officers and committees:

President—John Paul Jones.
Vice-President—D. L. Taze.
Secretary—C. M. H. Kaercher.
Treasurer—E. W. Gray.
Board of Governors—Philip Cohen, H. E. Wetzell, C. A. McKeeman.
Membership Committee—D. L. Taze, chairman; W. R. Moore, R. T. Southmayd, L. S. Ries, E. W. Gray, L. E. Slawson.
Auditing Committee—C. A. McKeeman, chairman; A. L. Vanderhoff.
Meetings and Entertainment Committee—H. P. Rankine, chairman; V. B. Kofoed, F. A. Kitchen.
Program Committee—Paul Gayman, chairman; W. M. Rowe, Philip Cohen.
Standards Committee—J. A. Schurman, Jr., chairman; H. E. Wetzell, J. E. Wilhelm.
Publication Committee—C. F. Eveleth, chairman; Walter Baggaley, D. K. Wright.
Legislation Committee—W. R. Beach, chairman; W. R. Rhoton.

A synopsis of meetings held during the year is given:

October 18, 1937. Subject: Lithium Chloride as a Dehumidifying Machine. Speaker: F. M. Johnson. Attendance: 63.

November 8, 1937. Subject: Corrosion—Its Causes and Control. Speaker: Ralph Baker. Attendance: 88.

December 13, 1937. Subject: Invading the Kingdom of Dust. Speaker: H. W. Tenney. Attendance: 89.

January 12, 1938. Subject: The Effect of Air Conditioning on Comfort, Health and Disease. Speaker: Prof. C.-E. A. Winslow. Attendance: 92.

March 14, 1938. Subject: Application of Motors and Motor Controls to Air Conditioning Equipment. Speakers: K. C. Mobarry and M. H. Russell. Attendance: 83.

April 5, 1938. Subject: Radiant Heating. Speaker: D. J. Purinton. Attendance: 60.

May 9, 1938. Subject: Demonstration of Latest Instruments Used in Air Conditioning. Speakers: Prof. G. L. Tuve and L. J. Seigel. Attendance: 107.

The practice of holding a dinner and business meeting preceding the presentation of the speaker has been continued. In addition a complete air conditioning problem was worked out by a committee and presented to the Chapter step by step at each meeting. This innovation was well received.

It is reported that the Chapter has 39 members with many others interested in Chapter activities and it is expected that the number of members will be greatly augmented during the coming year.

Wisconsin

The Wisconsin Chapter reports a total membership of 65 and activities will be in charge of the following Officers and Committees during the 1938-39 season:

President—Prof. D. W. Nelson.
Vice-President—H. C. Frentzel.
Secretary—T. M. Hughey.
Treasurer—A. S. Krenz.
Board of Governors—J. H. Volk, E. A. Jones and J. S. Jung.
Program Committee—Ernest Szekely, Chairman; W. A. Ouweneel, and F. W. Goldsmith.
Membership Committee—J. H. Volk, Chairman; C. A. Hays, and I. J. Haus.
Publicity Committee—H. W. Schreiber, Chairman; L. W. Hessler, and H. C. Frentzel.

Vice-Pres. H. C. Frentzel has prepared the annual report and meetings held during the 1937-38 season are summarized:

November 15, 1937. Dinner and meeting held at City Club. Speaker—John Novotny, chief engineer, University of Wisconsin Power Plant. Subject—Description of the central heating power house and the distribution system employed to heat all buildings on the Madison campus. Attendance—60.

December 20, 1937. Speaker—Prof. G. L. Larson, chairman, Department of Mechanical Engineering, University of Wisconsin. Subject—Movie of a canoe and fishing trip made in northern Ontario, Canada, during the previous summer. Technical session was an open forum in which everyone discussed general questions that came up during the year. Attendance—42.

January 17, 1938. Dinner and meeting held at City Club. Speaker—L. V. Teesdale, senior engineer, Forest Products Laboratory, Madison, Wis. Subject—Condensation in Building Walls. Attendance—90.

February 21, 1938. Dinner and meeting at City Club. Speaker—F. R. Bichowsky, consulting engineer, Dow Chemical Co. Subject—Comfort and Industrial Air Conditioning with Lithium Chloride. Attendance—65.

March 21, 1938. Dinner and meeting at City Club. Speaker—Dr. Lloyd Arnold, professor of bacteriology and public health, College of Medicine, University of Illinois. Subject—Influence of Air Conditioning Upon Public Health; Physiological Effects of Controlled Indoor Atmospheres and How People React to Changes in Their Air Environment. Attendance—48.

April 18, 1938. Dinner and meeting at City Club. Speaker—Prof. M. K. Fahnestock, University of Illinois. Subject—Air Conditioning Studies in the Research Residence constructed as a working laboratory for all phases of home cooling with special reference to producing comfortable conditions in summer at a low initial and operating cost. Attendance—45.

May 10, 1938. Dinner and Meeting at City Club. Following dinner members and guests were shown colored movies by Prof. G. L. Larson of his trip to the various Chapters when he was president of the Society. Speaker—E. Holt Gurney, Toronto, Canada, president of the A. S. H. V. E. Mr. Gurney told of his experiences in South Africa, describing the gold mine activities and benefits derived by air conditioning the mines. Attendance--61.

Pittsburgh

A complete report of activities of the Pittsburgh Chapter has been prepared by Secy. T. F. Rockwell and the following are the officers and committees who will serve for the 1938-39 season:

President—J. F. S. Collins, Jr.
Vice-President—R. A. Miller
Secretary—T. F. Rockwell
Treasurer—L. S. Maehling
Board of Governors—M. L. Carr, Chairman; E. H. Riesmeyer, Jr., and E. C. Smyers
Program Committee—E. C. Smyers, Chairman; H. A. Biber, R. H. Heilman, R. A. Miller, H. L. Moore, and John Proie
Social Committee—R. J. J. Tennant, Chairman; H. A. Canon, J. E. McLean, R. B. Stanger, and G. G. Waters
Student Committee—T. F. Rockwell, Chairman; F. C. Houghten, and B. D. Landes
Membership Committee—P. A. Edwards, Chairman; H. E. Park, E. H. Riesmeyer, Jr., W. W. Stevenson, and W. T. Van Alen
Publicity Committee—D. W. Loucks, Chairman; M. H. Jackson, A. W. Marshall, Leighton Orr, and G. W. Strachan.
Current Events—P. C. Strauch, Chairman; T. M. Dugan, Kermit Griest, and F. H. Hecht
National Activities—C. M. Humphreys, Chairman; F. C. Houghten, F. C. McIntosh, and T. F. Rockwell
Research Committee—F. C. McIntosh, Chairman; M. L. Carr, J. D. Edwards, F. C. Houghten, T. F. Rockwell, and F. N. Speller

Meetings were held from September through June and are summarized in the following report:

September 13, 1937. Forced Circulation Hot Water Heating, R. E. Moore, vice-president, Bell and Gossett Co. Attendance—47.

October 11, 1937. Forced Circulation Warm Air Heating, F. A. Kitchen, president, American Warming and Ventilating Co. Attendance—35.

November 8, 1937. Automatic Residence Heating with Bituminous Coal, R. A. Sherman, fuel engineer, Battelle Memorial Institute. Attendance—57.

December 13, 1937. Annual Dinner and election of officers. Attendance—40.

January 10, 1938. What Constitutes a Draft? Abstract of a research laboratory paper, by F. C. Houghten. Heat Transfer Through Single and Double Glazing, abstracted by Leighton Orr. Attendance—35.

February 14, 1938. District Heating, T. E. Purcell, superintendent, Allegheny County Steam Heating Co. Attendance—54.

March 14, 1938. Human Factors in Air Conditioning, Prof. B. M. Woods, University of California. Attendance—45.

April 11, 1938. Electrostatic Dust Precipitators, G. W. Penney, research engineer, Westinghouse Electric & Manufacturing Co. Attendance—65.

May 16, 1938. Code of Cast Iron Heating Boilers, J. F. McIntire, 1st vice-president, A.S.H.V.E. Attendance—54.

June 13, 1938. Golf, Dinner, Dance, South Hills Country Club. Attendance—70.

On September 1, 1938, the roll of membership of the Pittsburgh Chapter numbered 65.

Philadelphia

The annual report of the Philadelphia Chapter was prepared by Secy. H. H. Mather and at the present time there are 142 members on the chapter rolls.

The activities for the 1938-39 season are under the direction of the following officers and committees:

President—H. H. Erickson
Vice-President—R. F. Hunger
Secretary—H. H. Mather
Treasurer—C. B. Eastman
Board of Governors—Edwin Elliot, W. A. Bornemann, and L. P. Hynes
Meetings Committee—R. F. Hunger, Chairman; E. A. Dambly, P. L. Davidson, L. E. Moody, and C. S. Leopold
Membership Committee—Edwin Elliot, Chairman; E. H. Dafter, H. F. Dever, M. G. Kershaw, J. P. Murdock, Jr., E. C. Powers, Morris Sheffler, L. A. Tucker, and C. W. Whitney
Legislative Committee—M. F. Blankin, Chairman; W. P. Culbert, and C. S. Leopold
Finance Committee—C. B. Eastman, Chairman; H. G. Black, R. C. Bolsinger, L. P. Hynes, and W. F. Smith.
Year Book Committee—P. L. Pryibil, Chairman; R. S. Arnold, Lucien Buck, A. C. Caldwell, H. F. Dever, A. E. Kriebel, Stanley Plewes, and Karl Rugart
Special Advisory Committee—L. P. Hynes, Chairman; J. H. Hucker, and F. D. Mensing
Dinner Committee—A. H. MacDade, Chairman; H. C. Cody, and C. F. Dietz
Golf Committee—W. P. Culbert, Chairman; and R. C. Bolsinger

A summary of meetings held during the 1937-38 season is included:

October 14, 1937. Engineers Club. Speaker—Col. D. S. Boyden, president of the A. S. H. V. E., delivered a paper accompanied by slides on the Problem of Central Station Steam Distribution as Applied to the City of Boston.

November 11, 1937. Engineers Club. Speaker—Prof. C. P. Yaglou of the School of Public Health, Harvard University, spoke on The Effect of Air Conditions on Comfort, Health and Disease. His talk was illustrated with slides and pictures.

December 9, 1937. Engineers Club. Speakers—A. J. Johnson, director of the Anthracite Laboratories, Primos, Pa., delivered a talk on Coal; T. H. Smoot, chief engineer of the Anchor Fence Co., Baltimore, Md., spoke on Oil; and F. R. Trembly, Jr., Philadelphia Gas Works Co., discussed Gas.

January 13, 1938. Bookbinder's Restaurant. Social meeting with professional entertainers.

February 10, 1938. Engineers Club. Motion picture accompanied by sound on the subject of Heat and Its Control, presented by Johns-Manville Co. Speakers—N. A. Humphrey and L. C. Leslie on Selection of the Economic Thickness of Insulation.

March 10, 1938. Edward Bok Vocational School, Philadelphia. Inspection of this vocational school, particularly the mechanical equipment.

April 14, 1938. Engineers Club. Election of officers at annual meeting. Speaker—Thomas Urdahl, consulting engineer, Washington, D. C., on Air Distribution Problems in Air Conditioning.

May 11, 1938. Whitemarsh Valley Country Club—Annual Golf outing.

Southern California

The Southern California Chapter of the American Society of Heating and Ventilating Engineers reports a most successful year. The membership at the close of the year was 46 members.

For the season of 1938-39 the following officers have been selected to carry on the administrative work:

President—H. M. Hendrickson
Vice-President—J. F. Park
Secretary—A. J. Hess
Treasurer—W. D. Fabling
Board of Governors—E. L. Cawby, H. H. Douglas, E. H. Kendall, and R. E. Phillips.

The summary of the meetings held in 1937-38 is as follows:

September 1937. Dr. C. A. Mills, Professor of Experimental Medicine, University of Cincinnati, gave a talk on the Weather and Its Effect on the Human Being. In his talk he reported on his research activities using radiant surfaces for heating and cooling. Attendance—125.

October 1937. A. J. Hess, engineer with English & Lauer, made a talk on the subject of Modern Trends in Air Conditioning, followed by an active discussion by the members who were present. Attendance—35.

November 1937. Prof. L. M. K. Boelter, of the University of California, made a talk on The Fundamentals of Cooling Tower Design. Attendance—55.

December 1937. Prof. E. F. Dawson, of the University of Oklahoma, talked on the subject of Refrigeration and Air Conditioning. Attendance—65.

January 1938. Herman Seid, of the Auditorium Conditioning Corporation, gave a talk on the subject of Patented Systems and Refrigeration and Air Conditioning Ordinances. Attendance—80.

February 1938. F. G. Baender, of the Thermo Air Conditioning Institute, gave a talk on some Technical Aspects of Refrigeration, and a short discussion and demonstration of an evaporator. Attendance—65.

March 1938. Pres. E. Holt Gurney, of the American Society of Heating and Ventilating Engineers, delivered a very interesting talk on the General Status of the Society, and some interesting opinions on the General Aspects of the Air Conditioning Industry. Attendance—52.

April 1938. Verling Kersey, superintendent of Schools of the City of Los Angeles, gave a talk on The Development of a Technological Education. Attendance—50.

May 1938. Prof. B. F. Raber, of the University of California, gave a discussion on Fan Characteristics Useful in Air Conditioning, followed by R. C. Martinelli, of the University of California, who gave a demonstration of visual demonstrations of fan acoustics by means of the Cathode Ray Osfilograph. Attendance—82.

June 1938. The meeting was held at the Helms Bakery in Culver City near Los Angeles, where O. W. Ott, consulting engineer, delivered a paper entitled Package Air Conditioning. This was followed by a most interesting trip through the bakery, which is the largest air conditioned bakery in the world.

The year was quite successful in that the meetings were of great interest and of extreme educational value to all of those who attended.

It is of further interest that the local Chapter of the *American Society of Refrigerating Engineers* joined in several of the year's meetings.

The successful year just passed can be attributed to the effort of the outgoing officers consisting of:

President E. H. Kendall; Vice-President H. M. Hendrickson; Secretary J. F. Park; and Treasurer A. W. Cooper.

Western New York

The report of the Western New York Chapter of the Society was filed by Secy. W. R. Heath, and at the present time there are 78 members on the chapter rolls.

The following officers and Board of Governors were elected at the May meeting for the 1938-39 season:

President—J. J. Landers
First Vice-President—L. P. Saunders
Second Vice-President—C. A. Gifford
Secretary—W. R. Heath
Treasurer—H. C. Schafer
Board of Governors—M. C. Beman, B. C. Candee, Joseph Davis, Roswell Farnham, and D. J. Mahoney

The chairman of the Membership Committee is L. P. Saunders, and the chairman of the Program Committee is C. A. Gifford.

The following is a report of the activities of the Western New York Chapter for the 1937-38 season:

September 16, 1937. Golf meet at the South Shore Country Club, followed by dinner at the Au Revoir Restaurant. Attendance—35.

October 11, 1937. Dinner meeting at University Club. Speaker—L. A. Harding; Subject—The Sun. Attendance—55.

November 8, 1937. Dinner meeting at University Club. Speaker—R. E. Hattis; Subject—Heating and Air Conditioning of a Windowless Building. Attendance—42.

December 20, 1937. Social meeting at University Club. Attendance—50.

January 11, 1938. Dinner meeting at University Club. Speaker—Dr. C.-E. A. Winslow; Subject—The Physiological Reactions and Sensations of Pleasantness under Varying Atmospheric Conditions. Guest—Dr. Fronczak, Health Commissioner of Buffalo. Attendance—44.

February 8, 1938. Dinner meeting at University Club. Speaker—Prof. Philip Drinker; Subject—Atmospheric Pollution and Dust Control. Guest—Theodore Hatch, Dust Control Engineer of the Dept. of Labor, State of New York. Attendance—30.

March 14, 1938. Dinner meeting at Iroquois Gas Co. Speaker—W. B. Garrison, Jr.; Subject—Highlights in Connection with Supplying Gas for the New York World's Fair. Attendance—88.

April 25, 1938. Dinner meeting at University Club. A. S. H. V. E. officers and Council members were guests. Speakers—Pres. E. H. Gurney, on Air Conditioning Brings Human Problems to the Engineer; Vice-Pres. J. F. McIntire, on The Engineer; Treas. A. J. Offner; F. E. Giesecke, W. L. Fleisher, A. P. Kratz, C. W. Farrar, and Secy. A .V. Hutchinson. Attendance—80.

May 9, 1938. Annual meeting and election of officers at the University Club. Attendance—51.

A Study of the Heat Requirements of a Single-Glazed Test House and a Double-Glazed Test House

By M. L. Carr,[*] R. A. Miller,[**] Leighton Orr,[***] and David Shore,[†] *(MEMBERS)*

THIS paper reports the results of a study of the relative heat requirements necessary to maintain a temperature of 70 F in each of two similarly constructed test houses, one having single- and the other double-glazed windows. The investigation was planned and carried out by the Pittsburgh Plate Glass Co. with the aid and counsel of Pittsburgh Testing Laboratory during the period January 18 to April 22, 1933. The data were received for analysis in 1937 and were prepared for publication by the A. S. H. V. E. Technical Advisory Committee on Air Conditioning Requirements of Glass, consisting of the following members: M. L. Carr, Chairman, R. A. Miller, Vice-Chairman, F. L. Bishop, A. N. Finn, S. O. Hall, E. H. Hobbie, F. W. Parkinson, Harold Perrine, W. C. Randall, L. T. Sherwood, J. T. Staples, C. Tasker, G. B. Watkins and F. C. Weinert.

Test Houses and Apparatus

The test houses were erected on the top of a three-story laboratory building at Creighton, Pa. The houses with relation to their surroundings are shown in Fig. 1. The exterior appearance of one of the houses is shown in Fig. 2, while Fig. 3 shows the interior of one of the houses.

The interior dimensions of each house were: length and width, each 10 ft, 4¼ in.; height, 8 ft, 6 in. The walls of the houses extended up from the roof of the building to which they were anchored, the floor of each test house being 2 ft above the roof and level with the top of the building parapet walls. The houses were frame structures, sheathed and stuccoed on the outside, with 1 in. insulating board and gypsum plaster on the inside. One-fifth pitch gable roofs with cypress shingles laid on building paper were used in the construction.

Each house contained seven double-hung windows, each window containing two 26-in. by 30-in. lights of double-strength glass, and one wooden entrance door. The southwest, northwest, and northeast walls each contained two windows, while the southeast wall contained one window and the door. The spacing of the lights in the double-glazed windows was ⅜ in., and venting of the air between the space and the outside was provided. The vents were closed during these tests. The floor of each house was painted gray, while the walls and ceiling were painted white.

The percentage of glass area to other areas for the two houses were as follows:

Glass area to total heat transfer area	13.3 per cent
Glass area to total side wall area	21.5 per cent
Glass area to floor area	70.6 per cent

Heating Equipment

Each house was provided with a five-kilowatt electric unit heater as shown at the top of Fig. 3. The heating element of each unit was divided into two parts of three and two kilowatts, each of these having independent thermostatic control. The fan of each unit ran continuously, while the heating elements operated intermittently as the occasion demanded, being thermostatically controlled to hold the room at 70 F. The thermostats were located 42 in. above the floor in the center of the room and were protected from the sun's rays by a metal shield. The position of each unit was adjusted to give substantially the same rate of air movement over each window. The average velocity of the air over the inner faces of the lights was between 1.5 and 2 mph, as measured by a Kata thermometer.

Procedure

No attempt was made to build the houses unusually tight to avoid infiltration, but they were made as nearly equal in this respect as possible, and means were provided for checking this condition at intervals during the test. The method used was to apply a suction to each house and equalize the amounts of air drawn from each by sealing cracks. This was done at the beginning of the test and at monthly intervals thereafter.

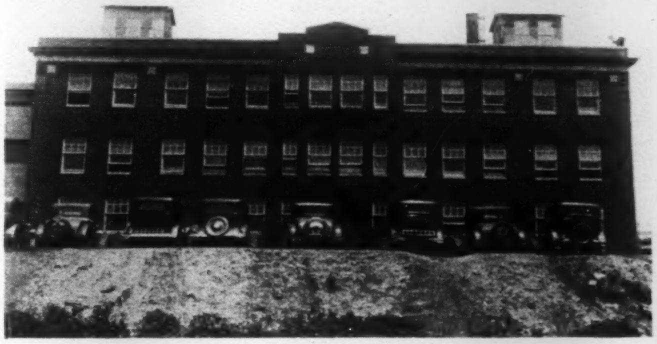

Fig. 1—Test houses on top of three-story laboratory building with relation to their surroundings

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[*] Director, Pittsburgh Testing Laboratory, Pittsburgh, Pa.

[**] Technical Sales Engr., Pittsburgh Plate Glass Co., Pittsburgh, Pa.

[***] Research Engr., Pittsburgh Plate Glass Co., Creighton, Pa.

[†] Junior Research Engr., A.S.H.V.E. Research Laboratory, Pittsburgh, Pa.

For presentation at the 45th Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., January, 1939.

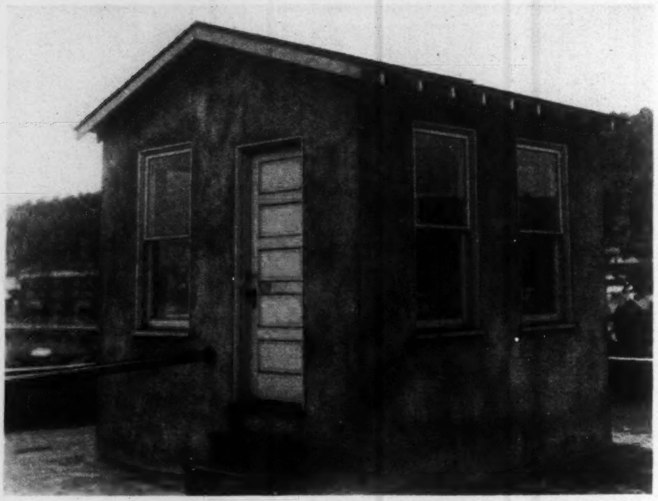

Fig. 2—Exterior of single-glazed test house. Double-glazed house identical

The electrical energy required to heat each house was measured with a recording watthour meter, which was read daily at 11:30 a. m. The time during which the two-kilowatt and the three-kilowatt elements in the units of each house were in operation was registered by recording wattmeters.

The temperature of each test house at a point near the thermostat and four feet above the floor was recorded. The average difference in temperature between the air at the center of the room and the inside surface of the window lights of each house was given by the thermocouples attached to the center of one of the lower window lights on each side of the house. These four thermocouples were connected in series, with their cold junctions in a reservoir at the center of the room. The thermocouple junctions were held against the glass surfaces with small drops of specially prepared cement.

Outdoor temperatures were recorded with a resistance thermometer, sheltered from the sun by a wooden housing, open only on the northeast side.

The wind velocity was measured midway between the test houses with a rotating cup anemometer coupled to an electric circuit and to a demand meter, utilized to record the number of revolutions of the cups in continuous five-minute periods. The device was calibrated with a rotating vane portable anemometer.

Sunshine intensity was recorded by means of a photronic cell connected to a recording millivoltmeter. The cell was attached to a disc driven by an electric motor, which caused it to follow the sun across the sky. The cell and motor were mounted outdoors under a protecting glass cover with the face of the cell normal to the sun's rays at noon. Provision was made for adjusting the inclination of the cell from time to time as the sun became higher in the sky with the advance of the seasons.

Test Results

The data obtained during the continuous 95-day test period were summarized by determining the average for each 24-hour period, beginning at 11:30 a. m., of the following:

(1) The daily average outdoor temperature was obtained by determining the average temperature for each hour from the recorded temperature chart, and averaging these 24 hourly averages.

(2) The temperature at the center of each house was taken from the recorder charts and did not vary much, being thermostatically controlled, except when the solar heat was more than sufficient to carry the heating loads.

(3) The daily average temperature of the inside glass surface for each house was computed, using a conversion from the millivolt recorder charts.

(4) The daily average wind velocity was obtained by dividing the total number of anemometer cup revolutions by the conversion factor.

(5) The sun intensity was determined from the area under the curve of the sunshine recorder by a planimeter, using the intensity at night as a base line. A square inch of sunshine area was arbitrarily taken as one unit of sunshine, and is so used in this report.

It should be emphasized that these values afford only a means of comparing one day with another with respect to sunshine, and do not give absolute values of solar radiation impinging on the various surfaces of the test houses.

A log of the daily average room temperatures, average inside glass surface temperatures, weather conditions and average daily electrical heat requirements of the test houses for the test period is plotted in Fig. 4. Also included in this plot are the calculated heat requirements of the two houses based on the accepted transmittances and film conductance coefficients of the materials used, and infiltration losses, assuming no solar heat input. The ratio of the electrical heat supplied to the houses also appears in this Figure. The average daily electrical input in Btu per hour is plotted against temperature difference between the inside and the outside in Fig. 5. The relationships between the average daily electrical heat supply in Btu per hour per degree Fahrenheit and the mean temperature, the wind velocity in miles per hour, and the sun intensity in units are plotted in Figs. 6, 7 and 8, respectively.

The relationship between the total daily heat supply in Btu per hour per degree Fahrenheit (electrical heat plus estimated solar heat) and the wind velocity in miles per hour is shown in Fig. 9.

The relationships of the ratios of the electrical heats supplied to the houses to the temperature difference between inside and outside, the wind velocity and the sun intensity, are given in Fig. 10.

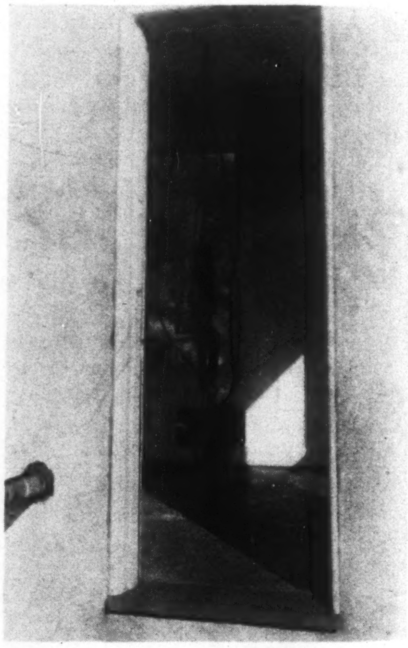

Fig. 3—Interior of single-glazed test house. Interior of double-glazed test house identical

Discussion of Results

An examination of Fig. 4 will serve to substantiate the following generalities:

(1) The test houses were maintained at substantially 70 F at the center of each house except for a few times when this temperature was exceeded due to sun effect.

(2) The 24-hour average outdoor temperatures varied from 7 F to 65 F during the test period. The actual temperature extremes were 0 F and 82 F.

(3) The 24-hour average wind velocities varied from 1.5 mph to 14 mph. The extreme velocities were 0 mph and 32 mph.

(4) The average temperature of the inner surface of the glass in the single-glazed house remained about half-way between the inside and the outside temperatures, with little or no sunshine and wind, while the temperature of the inner surface of the inside lights of the double-glazed house was about one-quarter of the way between the inside and outside temperatures under the same conditions, being closer to room temperature.

Comparison of Measured and Calculated Heat Requirements

The daily variation of the measured electrical heat input in Btu per hour for both houses is shown in Fig. 4, together with the calculated heat requirements of the two houses based on the prevailing outside temperature and wind velocity and the accepted heat transfer constants taken from the A.S.H.V.E. Heating, Ventilating, Air Conditioning Guide, no allowances being made for the sun effect. The effect of infiltration is also included in this calculation. The calculated curve is higher in most cases than the measured curve, as is to be expected, since the inclusion of solar heat in the calculated curve would lower its value. It should be noted that the differences between the two curves, generally speaking, are greater on days of more intense solar radiation.

Variation of Heating Requirements with Weather Conditions

The variation of the average daily electrical input in Btu per hour with temperature difference between the inside and outside for both houses is shown in Fig. 5. A lack of sufficient points in the low temperature difference range makes it rather difficult to predict this part of the curve. The two curves do not intersect the temperature difference axis at zero, indicating that sufficient amounts of heat were stored in the houses to keep them at 70 F when certain outdoor temperatures less than 70 F were attained. The curve for the single-glazed house intersects the temperature difference axis at about 2.5 F, while the one for the double-glazed house intersects it at about 5.5 F. This indicates that for these respective temperature differences no artificial heat was required to maintain the predetermined control temperature of 70 F. The fact that no heat was required for the double-glazed house at a temperature of about 64.5 F as compared with about 67.5 F for the single-glazed house indicates that the double-glazed house retained heat more effectively than did the single-glazed house.

The variation of electrical heat supplied to the houses in Btu per hour per degree Fahrenheit with mean temperature is shown in Fig. 6. For both the single- and double-glazed houses there was an apparent increase in the number of Btu per hour per degree Fahrenheit

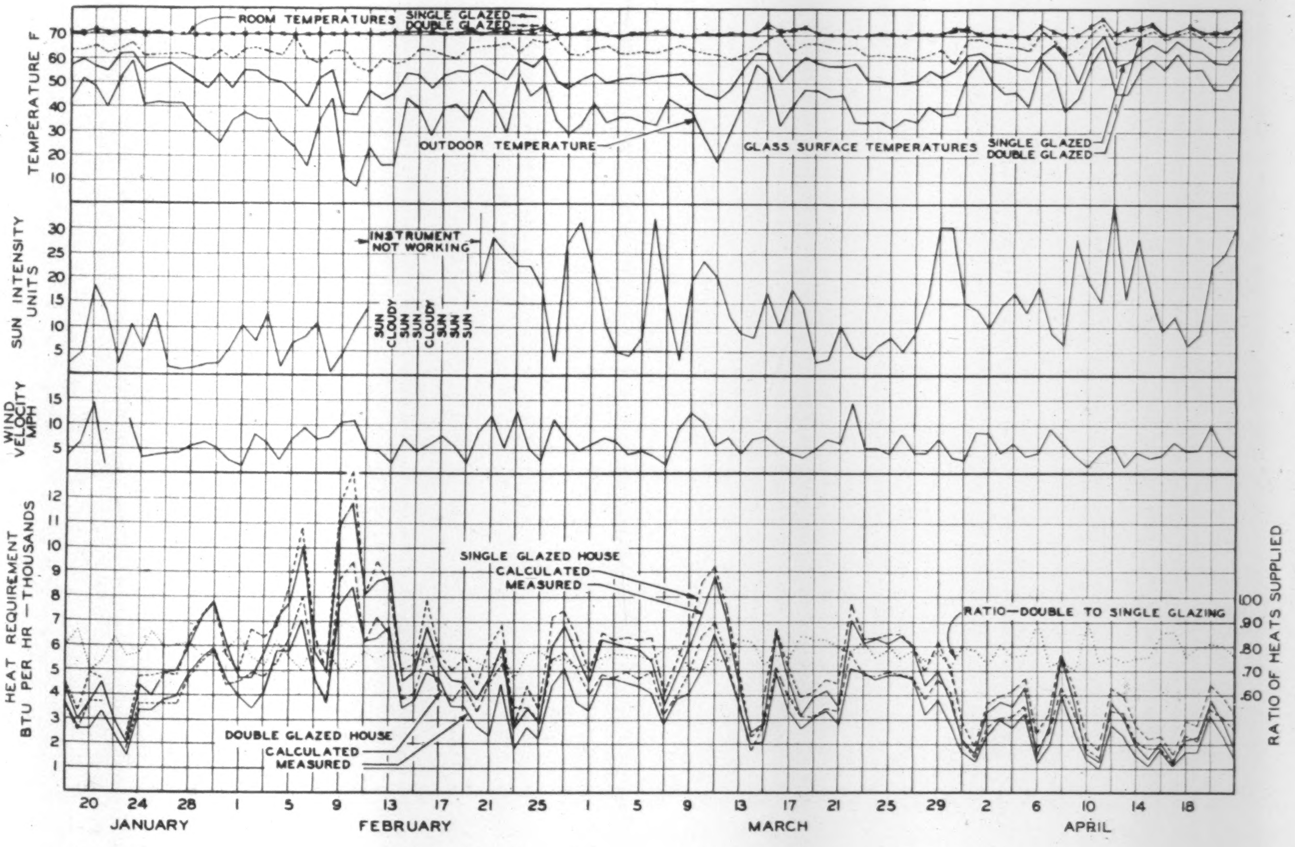

Fig. 4—Log of test data—January 18 to April 22, 1933

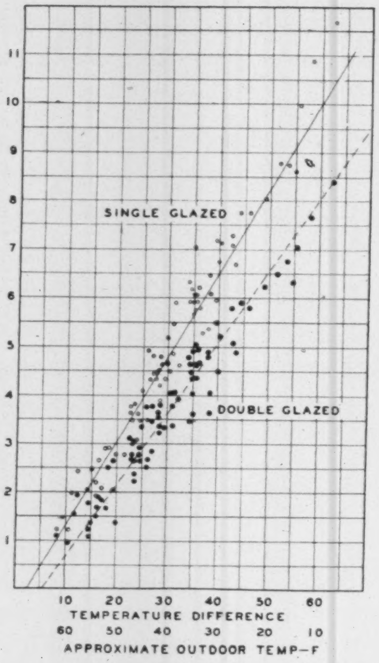

Fig. 5—Relation between mean temperature and electrical input in Btu per hour for single- and double-glazed houses. Approximate outdoor temperature based on an assumed indoor temperature of 70 F

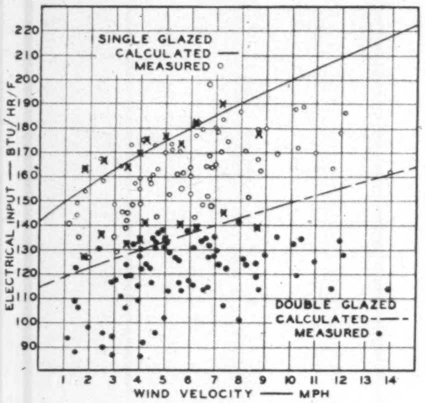

Fig. 7—Relation between wind velocity in miles per hour and electrical input in Btu per hour per degree Fahrenheit for single- and double-glazed houses. Curves giving calculated requirements based on prevailing temperature difference and wind velocity and GUIDE transmission coefficients. No allowance for sun radiation. *X* points represent electrical input on days with little or no sunshine

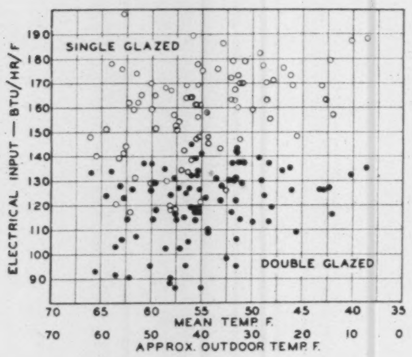

Fig. 6—Relation between mean temperature and electrical input in Btu per hour per degree Fahrenheit for single- and double-glazed houses. Approximate outdoor temperature based on an assumed indoor temperature of 70 F

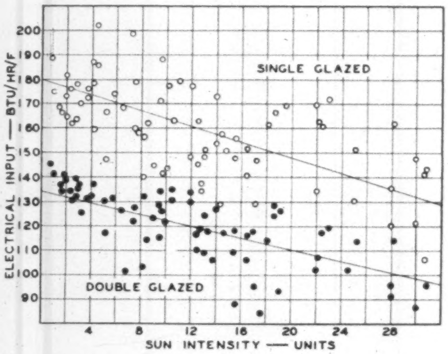

Fig. 8—Relation between sun intensity and electrical input in Btu per hour per degree Fahrenheit. No allowance for wind velocity

required with decreasing mean temperature, and therefore also with an increasing temperature difference between the inside and the outside. This is contrary to the accepted fact that the heat transfer through any material per degree temperature difference increases with increasing mean temperature. It is also contrary to the generally accepted belief based on a number of surveys that the heating requirements of a building per degree temperature difference between the inside and outside decreases with increasing temperature difference, or in other words that the efficiency of heat utilization increases with the heat demand. A possible explanation, at least for a part of this difference, is that in these tests there was a continuous thermostatically controlled electrical heat supply which operated only when the occasion demanded it. Thus, all of the heat supplied by the electrical heater was necessary to maintain the controlled temperature, and this required amount of heat per degree temperature difference decreased as the mean temperature approached that critical temperature at which no artificial heat was required. On the other hand, in the case of a home or an office building, where the occupancy requires heat only during certain hours of the day, a usual occurrence is the supply of more than sufficient heat, especially on warm days. This overheating raises the Btu per hour per degree Fahrenheit supplied above that necessary, causing a condition of less efficient heating at higher outdoor temperatures. The data plotted in Fig. 6 take no cognizance of sun radiation, wind velocity or infiltration. Infiltration may account for the indicated increase in heat demand per degree temperature difference for lower mean temperatures. This would not be the case if the volume of infiltering air were constant over the range of mean temperatures. However, since temperature difference or the *chimney effect* in a building serves as a motivating force to increase infiltration in addition to the wind effect, this may serve to produce the observed effect.

The variation of electrical heat supplied in Btu per hour per degree Fahrenheit with wind velocity in miles per hour is shown in Fig. 7. The points for both the single- and double-glazed houses indicate a definite increase in heat demand with increasing wind velocity. The calculated curves for the two houses are also shown, based on the assumed transmittance and surface conductance coefficients for the building materials used in the constructions, and coefficients of 0.88 to 1.42 and 0.51 to 0.66 for the single- and double-glazed windows, respectively, the variation being caused by a 0-15 mph range in wind velocity. These curves include heat input necessary due to infiltration losses. The scattering of points away from the calculated curve results from neglect of the solar radiation effect. Thus, the sunnier the day, the farther the heat input points are from the calculated curve. The points follow the general directional trend of the calculated curve for velocities up to about eight miles per hour. For higher values of velocity, the points indicate a decreasing effect of velocity. This bears out earlier A.S.H.V.E. Laboratory studies,[1] which indicate that the effect of wind velocity in increasing heat requirements of buildings may be over-emphasized in present calculation methods. The *X*-points are for days when there was little or no sunshine. For the single-glazed house these *X*-points conform quite well to the calculated curve; however, for the double-glazed house, with one exception, they lie above the calculated curve. This would indicate that the value of 0.51 to 0.66 used for the transmission coefficient in the double-glazed case may be too low.

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[1] Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 387.)

The relationship between the electrical heat supplied in Btu per hour per degree Fahrenheit and sun intensity in units is given in Fig. 8. For both the single- and double-glazed houses there was a definite decrease in the amount of electrical heat required with increasing sun intensity. The scattering of the points was due, in part, to the wind velocity effect of which no account was taken in this plot.

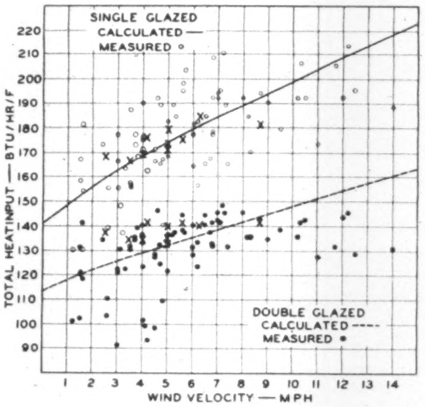

Fig. 9—Relation between wind velocity in miles per hour and total heat input (electrical + solar heat) in Btu per hour per degree Fahrenheit for single- and double-glazed houses. Curves giving calculated requirements based on prevailing temperature difference and wind velocity and GUIDE transmission coefficients. *X* points represent electrical input on days with little or no sunshine

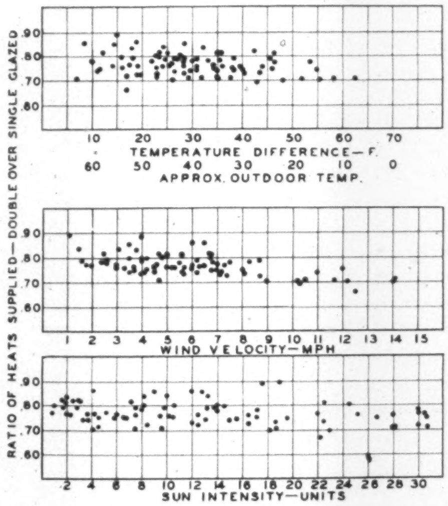

Fig. 10—Relation between temperature difference, wind velocity and sun intensity and the ratio of the heats supplied to the houses (double- over single-glazed)

An attempt was made to combine the wind velocity and sun intensity effects by adding to the measured heat supply an amount of heat proportional to the sun intensity for any test day. This procedure should tend to eliminate the sun effect for any given day, and thus give the value of heat required under a condition of no solar radiation. The result of this attempt is shown in Fig. 9 in which the measured electrical input plus 1.55 times the solar intensity for the single-glazed house and 1.10 times the solar intensity for the double-glazed house is plotted against the wind velocity. The *solar intensity multipliers* represent the negative of the slopes of average curves drawn through the points of Fig. 8. On this basis the points plot better, tending to arrange themselves about the calculated curves for the single- and double-glazed houses. These calculated curves are the same as those which appear in Fig. 7, while the *X*-points again indicate days when there was either little or no sunshine.

The variation in the ratio of the heat required for the double- and single-glazed houses with temperature difference between the inside and outside, the wind velocity and the sun intensity is shown in Fig. 10. Heat loss from either building should be proportional to the temperature difference multiplied by the integrated transmittance coefficients for the ceiling, the floor and the walls, including the windows. Since such integrated coefficients must vary with single and double glazing, the ratio between the heat losses by conduction through the structure should be constant for any temperature difference, and, therefore, other factors being equal, the ratio between the heat requirements of the double- and single-glazed houses should be constant, insofar as it is affected by transmittance through the walls. This, however, is not shown to be the case in Fig. 10 (top plot), which indicates a higher ratio for lower temperature differences. This variation may be due to the effect of wind and sun intensity. There is an apparent greater dispersion of the points upwards for low temperature differences, with a constant value of approximately 0.70 for the lower limit. It may be that this lower ratio of 0.70 is indicative of the transmittance effect, and the upward dispersion of the points the effect of wind velocity and sun intensity.

Wind velocity should have two effects on the ratio between the heat required for the double- and single-glazed houses. Other things being equal, infiltration into the two houses should be independent of single or double glazing, and, in fact, in the two houses under consideration an attempt was made to make infiltration equal. Hence, an equal increase in heat loss by infiltration for both houses independent of glazing should tend to increase the ratio with increased wind velocity. Increase in wind velocity also increases transmission through windows by increasing the outside film conductance coefficient. Since the outside film conductance coefficient for the double-glazed window is a smaller proportion of the combined effect for the entire window than in the single-glazed case, the increased wind velocity is accompanied by lesser increase in transmission through the double-glazed than through the single-glazed window. This effect should decrease the ratio with increasing wind velocity. Evidently the latter effect was greater than the former since the ratio decreases with increasing wind velocity as shown by the points of Fig. 10 (middle plot).

The effect of sun radiation should be the same in reducing the heat requirements for the two houses, insofar as all forms of construction other than the glass are concerned. Either single or double glazing has a relatively small effect in intercepting solar radiation through the windows. From the best data available,[2] each thickness of glass may be expected to intercept approximately 10 per cent of solar radiation. This means that approximately 90 per cent of the solar radiation incident against a window should pass through the single glazing, and 81 per cent through double glazing. Hence, it would be expected that the effect of sun radiation should be to increase slightly the ratio of the heats supplied. On the other hand, a double-glazed window is a better *heat trap* than a single-glazed window. Consequently, while slightly less heat can get in by direct radiation, considerably less can get out by transmission. The bottom plot of Fig. 10 shows a slight decrease rather than increase in the ratio of the heats supplied with increasing sun effect, indicating that the greater *trapping* effect of the double glazing overbalances its greater absorption.

Conclusions

1. By reason of the fact that under the same conditions the inside surface of the glass in the double-glazed house will be warmer than the same surface in the single-glazed house, occupants of a double-glazed house will feel more comfortable than those in a single-glazed house. This fact also means that frost will form more readily on the inside surface of the glass in the case of a single-glazed house than it will in the case of a double-glazed house.

2. The double-glazed test house described required from 70 to 80 per cent as much heat as the single-glazed test house. These ratios were affected somewhat by wind velocity, temperature difference and sun intensity, the ratio decreasing with increase in wind velocity and sun intensity, and with an increase in temperature difference.

3. The values shown for the ratio of the heats supplied to the test houses would apply only for the ratio of wall to window area and for the particular construction of these test houses. Had either this ratio or the construction been different, then the ratio of the heats supplied would also have been different. As the ratio of window area to wall area increases the ratio of the heats supplied to the two houses would decrease, the limit being the case of an all-glass house, in which case the ratio would be about 50 per cent. Likewise, if the ratio of window area to wall area is held constant, and the construction so changed as to increase the insulating value of the walls and roof to a theoretical case of no heat flow through them, then the ratio of the heats supplied would again be about 50 per cent, since all the heat would have to flow through the glass.

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[2] Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten, (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 93.)

Atlanta Chapter Plans for Active Year

October 4, 1938. Pres. C. L. Templin called the Atlanta Chapter meeting to order in the Biltmore Hotel at 8:00 p.m. Roll call revealed approximately 50 per cent of the membership present and the minutes of the last meeting were approved.

Reports were called for by President Templin and Chairman W. J. McKinney of the By-Laws Committee, suggested that each member of the Chapter be furnished with a copy of the existing By-Laws for study prior to discussion at the next regular meeting. This suggestion was unanimously accepted and Secy. T. T. Tucker was instructed to mail copies to all members.

Vice-Chairman C. E. Brockinton of the Membership Committee reported that four prospective applicants were under consideration and it was anticipated that their formal applications would be received at an early date.

A written report of a meeting held by the Codes Committee was submitted by Chairman W. P. Sherman.

C. T. Baker outlined plans for the Foods Preservation Conference at Knoxville, Tenn., October 20-21, giving a general outline of the history and purpose of this conference. It was announced that Messrs. Baker and Tucker expected to drive to the conference and that any of the members wishing to attend could make arrangements with them for transportation.

An interesting talk on Selling Engineering was presented by P. C. Stone, of Atlanta Gas Light Co. Quoting from his experience as supervisor of sales, Mr. Stone stated that without excellent engineering, he would have little to offer the purchaser. However, Mr. Stone expressed the belief that to successfully merchandise engineering, it is necessary to confine selling to simple language which can be readily understood by the ultimate consumer.

The next item on the program was a discussion of How to Figure Residence Heating by F. L. Laseter, Atlanta Gas Light Co. Typical estimate sheets were distributed for the convenience of members. Mr. Laseter stated that in approximately 30,000 surveys of residence heating completed in greater Atlanta, only about 3,000 plants were found large enough to satisfactorily meet requirements. This condition was attributed to many causes, but Mr. Laseter maintained that only through well directed codes, approved by the Chapter, would it be possible to eliminate a condition of this kind. Keen interest in the subject matter was shown by the discussion which followed, and comments were made by Messrs. R. L. Kent, C. T. Baker, M. G. Driscoll, Copeland, and others.

September 13, 1938. Members and guests of Atlanta Chapter met at the Biltmore Hotel and were called to order by Pres. C. L. Templin at 8:00 p.m. After roll call the minutes of the previous meeting were read and approved.

President Templin formally notified the Chapter of the transfer and resignation of H. K. McCain, secretary-treasurer.

The growth of the Chapter from its organization less than two years ago was outlined by President Templin, who expressed his hope for continued future expansion. With this in view he called the meeting's attention to the fact that the officers in the past had been required to perform all of the services necessary and pertinent to carrying on Chapter business. This had been previously discussed at a special meeting of the Board of Governors on September 7 and President Templin outlined a plan, which they had approved, whereby a greater number of the members would be given an active part in affairs of the Chapter. It was proposed that the By-Laws be amended to meet the Chapter's needs.

The office of Secretary-Treasurer was divided and nominations were called for from the floor. By unanimous acclaim T. T. Tucker was elected secretary and C. B. Cole, treasurer.

A series of standing committees to be appointed by the president was then proposed and discussed, and the duties of each committee were outlined. Committees were appointed as announced in the annual report of the Chapter.

A general discussion from the floor covering various programs and entertainments for the coming season was the next order of business. It was decided, however, that this matter should be left in the hands of the Entertainment and Program Committee. Some of the suggestions made were: (1) more papers by local Chapter members; (2) a question box; (3) expressions from the floor by members on subjects of particular interest; (4) professional or outside speakers; and (5) pepper-uppers.

The Permeability of Building Materials to Water Vapour

By J. D. Babbitt,* Ottawa, Canada

THE problem of the condensation of moisture in buildings has recently been discussed in two papers[1, 2] and the general nature of the problem has been outlined. The authors of the papers are in agreement on all points and have reached certain definite conclusions with regard to the causes and treatment of the trouble. They conclude from their studies that the moisture comes from the warm side of the wall (in ordinary buildings, the interior; in refrigeration, the exterior); that it migrates through the wall until it reaches a point where the temperature is below the dew-point, and there it condenses, giving rise to free water and in some cases to ice. This accumulation of water and ice not only destroys the insulation of the wall to a large extent but it also leads to structural deterioration from dry rot, etc. Under certain conditions, however, it has been found that there is no condensation of moisture even though the temperature is well below the dew-point corresponding to the temperature and humidity within the building. There are two conditions under which this desirable state exists: first, the moisture may be prevented from diffusing through the wall by an impervious barrier at the interior surface; or secondly, there may be an easy path for the water vapour to continue its migration towards the exterior. The vapour pressure tends to decrease with lowering of temperature so that the vapour always tends to move to the coldest point. It is only when this diffusion is restricted in some way that an accumulation of moisture occurs.

Whether or not there will be condensation in a wall, and the point at which it will arise, is dependent on the following factors:

1. The humidity within the building.
2. The temperature gradient in the wall.
3. The relative ease with which water vapour can pass through the various sections of the wall.

When the temperature at some point in the wall is below the dew-point and if that portion of the wall on the warm side of this point allows the passage of water vapour more easily than the portion of the wall on the cold side, there will be condensation. If the warm side is less permeable than the cold side there will be no condensation. Both of the previously mentioned papers[3, 4] stressed this fundamental point and, to alleviate this condition, it was suggested that some vapour-proof barrier be used on the interior portion of the wall combined with ventilation to the exterior. The ventilation may be dispensed with if the vapour barrier is sufficiently efficient and, generally, there is sufficient leakage through cracks and joints for ventilation without the addition of special openings for this purpose. Rowley and Teesdale emphasize the point, which was previously overlooked, that the exterior portion of the wall should not be made vapour proof but that all vapour barriers should be on the interior.

It appears then from these studies that the condensation of water in a wall depends, among other factors, on the permeability of the wall to water vapour, or rather on the relative permeability of the interior and exterior portions of the wall. This point has been vividly expressed by Rogers[5] by means of an illustration showing two basins into which water is running at the same rate. The accumulation of water depends on the relative size of the outlet to inlet. So in a wall the accumulation of moisture is dependent on the relative permeability to moisture of the two portions of the wall.

In order to plan a wall construction which will be proof against moisture condensation it is necessary, therefore, to insure that less water vapour will diffuse into the wall from the warm side than can be carried away by diffusion towards the exterior. In planning such a wall it is essential to know (1) the properties of the various building materials with regard to water vapour transmission and (2), the actual construction of the wall taking into consideration such details as cracks, joints in the material and openings purposely left for ventilation. It is difficult to assess the importance of this second problem depending as it does on the construction of the building and, of course, almost entirely on individual workmanship. In the case of a material like wood sheathing it is probable that, although there is a considerable transfer through the wood itself, by far the greater part of the moisture diffuses through the cracks between the boards. There is need for an investigation on the amount of vapour diffusion through cracks of various sizes. Data on the best methods of sealing the joints between sections of building paper are essential before any real estimation can be made of the permeability of a structure to water vapour. Rowley[6] has initiated studies on this aspect of the subject by comparing the efficiency of building paper as a vapour stop under two conditions: (1) the joints in the paper were lapped only; and (2) they were lapped and sealed. In the second case slightly less moisture passed through the structure, indicating that there is some leakage through lapped joints. It is impossible from the available data to estimate the importance of the joints and

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[*] Division of Physics and Electrical Engrg., National Research Council.

[1] Condensation Within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund. (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1938, p. 49).

[2] Condensation in Walls and Attics, by L. V. Teesdale (Forest Products Laboratory, Madison, Wis., October, 1937.)

[3] Loc. Cit. See Note 1.

[4] Loc. Cit. See Note 2.

[5] Preventing Condensation in Insulated Structures, by T. S. Rogers. (*Architectural Record*, 83, 109, March, 1938.)

[6] Loc. Cit. See Note 1.

cracks existing in the wall compared to the leakage through the material itself and more quantitative work will have to be done before satisfactory estimates can be made. Rowley has also investigated the effect of boring four 1-in. vent holes in the outer side of the wall and found that under these conditions no condensation was obtained. It will be interesting to learn what is the minimum size of hole that is required to ventilate a given air space under specified conditions, but greater knowledge is required before this question can be answered.

It is very desirous in estimating the permeability of any wall to water vapour to know the vapour transmission coefficient of the various materials. Unfortunately at the present time this datum is available for few building materials. Both Rowley and Teesdale were interested in a general elucidation of the problem of condensation in walls and they were concerned with discovering the general principles governing the phenomenon rather than the properties of the individual materials. Rowley gives values showing the weight of condensate obtained with various walls under the test conditions and thus gives a means of comparing the relative efficiency of the various wall constructions observed. However, it is impossible to compare these results with measurements obtained in other laboratories under slightly different conditions. A knowledge of the absolute values of vapour transmission coefficients of all building materials is essential. It is suggested that a table of diffusion coefficients be prepared similar to the table of thermal conductivities published in the A.S.H.V.E. HEATING, VENTILATING, AIR CONDITIONING GUIDE. When such a table is available it will be possible to calculate the permeability of a structure to water vapour in a manner similar to that used for determining the thermal transmission of a wall computed from the conductivity values of the various building materials.

Explanation of Diffusion Coefficient

Before discussing the methods by which it is possible to obtain an absolute value of the diffusion coefficient, an examination of the meaning of this term should be observed more closely. In a literal interpretation the term *diffusion* is used to define the phenomenon by which one fluid (liquid or gas) is able to mix with another when brought into contact with it. This phenomenon is due to the movement of the molecules themselves, which by means of the energy of thermal motions tend to distribute themselves systematically throughout the mass of fluid. In the case of the diffusion of water vapour through building materials, the movement of a gas through a solid is of importance, rather than the movement of one gas into another. This is further complicated by the fact that there is definite evidence that in many cases it is not a diffusion phenomenon at all. In diffusion there should be no reaction between the two types of molecules, but only elastic collisions. In the transfer of water vapour there is in many cases adsorption at one surface, migration of the adsorbed particles along internal surfaces, and liberation at the other surface. This results in a net transfer of water, but the relation between the water and the material is much more intimate than in true diffusion. Also there is no relation between the permeability of a substance to air and its permeability to water, showing that the transfer of moisture is a different process. Rubber has been shown to be 50 times as permeable to water vapor as to hydrogen and, since lighter gases diffuse much more rapidly than heavier ones, this indicates that there is some additional action in the case of water.

However, in treating the transfer of water vapour through solids it is more satisfactory to assume that it is a diffusion process and thus to use the relations which have been previously established. What measurements have been made indicate that within broad limits the diffusion formulae give satisfactory results. Accordingly, Fick's Law can be used, which can be stated as follows:

$$W = \frac{dAt}{X}(P_1\text{-}P_2) \quad \ldots\ldots\ldots\ldots [1]$$

where, W = weight of water vapour diffusing in time t through a material of thickness X and area A, when the vapour pressures on the two sides of the material are P_1 and P_2 respectively. The coefficient of diffusion or the diffusivity is d. It is interesting to note that Equation 1 is exactly similar to Fourier's Law for the transfer of heat through a homogeneous material if Q, the total heat transferred, is substituted for W; k, the coefficient of conductivity, for d; and T_1 and T_2, the temperatures, for the pressures P_1 and P_2. If then the vapour migration obeys Fick's Law and the coefficients d are known, water vapour diffusion can be handled in a manner exactly similar to the treatment of heat transfer.

From this formula the diffusivity d can be defined as the weight of water vapour passing through a unit of area of the material when a unit of vapour-pressure difference is maintained per unit of thickness. Unfortunately there are several units which may be used to express d. Using the metric system the most convenient unit of measurement is grams per 24 hours per square meter of area with a vapour-pressure difference of one millimeter of mercury per centimeter thickness. In the English system of units the expression is pounds per 24 hours per square foot of area with a vapour-pressure difference of one inch of mercury per inch of thickness.

It is convenient to define the diffusance D (analogous to the conductance in heat transfer) as the weight of water vapour passing through a unit of area when a unit of vapour-pressure difference is maintained between the two surfaces of the material. In a manner similar to the treatment of heat transfer the resistivity and the resistance can be defined as the reciprocals of the diffusivity and diffusance respectively. In calculating the resistance of a built-up wall to water vapour it is only necessary to add the resistances of the individual sections as in the case of heat transfer.

There are two important considerations to be investigated with respect to water vapour migration. The first is to determine how reliably Fick's Law represents vapour transfer and the second is to evaluate d for all materials used in wall construction. The investigations on building materials which are at present available are practically negligible, for with the exception of some measurements on wood[7, 8] and a few building papers, there have been no measurements of the diffusion coefficients of building materials. There have been, however, numerous investigations on the vapour permeability of other classes of materials so that there already is a fairly respectable mass of data on which to base some preliminary conclusions. A paper by Carson[9] summarizes previous measurements and from these observations some generalizations may be established. In the following discussion of the measurement of the diffusion coefficients of building materials the conclusions are based on the findings given by Carson.

The Measurement of the Diffusion Coefficient

Assuming that the transfer of water vapour through a solid material obeys Fick's Law it is a simple matter to evaluate the coefficient *d*. Carson has listed five methods that have been used. The simplest method and the one most commonly employed is to use a dish or cell which can be sealed off by means of the material whose coefficient is to be measured. Distilled water or an aqueous solution of required vapour pressure is put in the cell. It is then placed in a constant temperature enclosure where the humidity is maintained at a low value by means of some desiccant. The loss of moisture through the specimen can then be determined by weighing at successive intervals. When an equilibrium condition is reached, *d* can be calculated from Fick's Law, since all the necessary data are known. A certain time is required before the loss of moisture becomes uniform owing to the fact that most building materials are hygroscopic and there is a definite change of moisture content with humidity. If the weight of the cell is plotted against time, a linear relation will be obtained when equilibrium has been reached. The slope of this line gives the loss of moisture per hour. This method can also be reversed in that the desiccant can be placed in the cell and the enclosure maintained at high relative humidity. The cell which is used in this method may be either a crystallizing dish in which case the specimen fits in flush with the top of the dish, or it may be a more complicated cell in which the specimen rests on a flat horizontal surface. The two arrangements as used in the National Research Laboratories are shown in Figs. 1 and 2. In both cases the validity of the results is dependent on the efficacy of the seal between the specimen and the vessel.

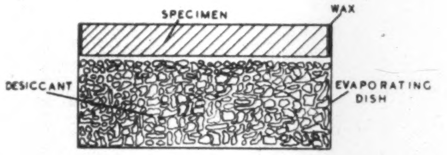

Fig. 1—Evaporating dish for testing water vapour transmission through specimen

Even when Fick's law is accepted as valid there are several precautions which must be observed in making such a measurement. Carson's study has shown a very wide range in the results obtained by different investigators on the same material. Some of these results may be accounted for by the variation in the material itself, but undoubtedly the greater part is due to a lack of proper precaution in making the measurements. The sources of error which are most apt to destroy the value of the measurements are the following:

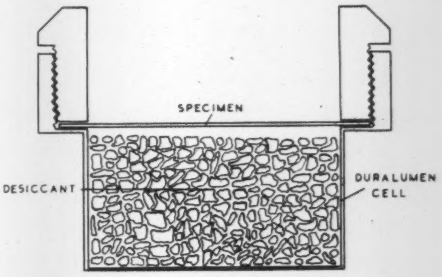

Fig. 2—Duralumin cell for determining water vapour transmission through specimen

1. *Leakage:* Unless the specimen is sealed tightly to the vessel, there is the possibility of moisture escaping from the cell without passing through the test material. Different substances have been used by different investigators to prevent leaks and such materials as beeswax, beeswax and rosin mixture, rubber solution, petrolatum, shellac and various other sealing waxes have been reported satisfactory. The most convenient method by which to test the efficiency of a material as a seal is to use it with some material such as a metal plate which allows no vapour to pass through it. In this case any loss of weight would be through the sealing compound. Irrespective of the nature of the sealing substance great care must be taken with each individual specimen as there is no method by which a leak can be discovered except by repeating the measurement. Carson, in studying the results of different investigations, came to the conclusion that there is a relation between the value of the coefficient obtained and the amount of care taken by the investigator to seal the sample.
2. *Edge Losses:* Such losses arise through a lateral flow of water vapour in the specimen with subsequent loss through the cut edges to the exterior. It is much more likely to occur in a cell of a type shown in Fig. 2 than in Fig. 1. In all cases, however, the cut edges of the specimen should be sealed so that all possibility of such error is eliminated.
3. *Vapour-Pressure Difference Across the Specimen:* An element of doubt exists in regard to the accuracy of the measurements of the vapour pressures at the two faces of the sample. Since air itself offers a finite resistance to the transfer of vapour, it is wrong to suppose that the vapour pressure at the surface of the material is the same as that at the surface of the liquid, unless the liquid is very close to the surface or the rate of transfer of water vapour is very slow. In other words, unless the resistance of the air space is negligible compared to the resistance of the specimen, it is not justifiable to assume that the vapour drop across the specimen is equal to the overall pressure drop. The pressure difference between surface of liquid and surface of material depends on the distance between these two points and on the amount of water vapour that is being transferred. The total pressure drop between aqueous solution and desiccant is divided into three separate terms depending on the following resistances:

(a) Resistance of air space between surface of aqueous solution and surface of specimen.
(b) Resistance of specimen.
(c) Resistance of air space between surface of specimen and desiccant.

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[7] The Movement of Moisture with Reference to Timber Seasoning, by S. T. C. Stillwell (Forest Products Research, Dept. Sci. Ind. Res., Great Britain. Technical papers No. 1, 1926.)

[8] Moisture Movement Through Wood: The Steady State, by J. F. Martley. (Forest Products Research, Dept. Sci. Ind. Res., Great Britain, Technical papers No. 2, 1926.)

[9] Permeability of Membranes to Water Vapour with Special Reference to Packaging Materials, by F. T. Carson. (Bureau of Standards, Miscellaneous publication M127, August, 1937.)

The relative magnitudes of these three terms are the deciding factors in the distribution of the pressure gradient. If the resistance of the specimen is large compared to the combined resistances of the air spaces, then the greater part of the vapour pressure drop will be across the specimen and no sensible error will be introduced by using the overall pressure drop in the equation to determine *d*. On the other hand when the resistance of the specimen is so small that it is comparable to the resistances of the air spaces, then only a fraction of the pressure drop is due to the specimen and allowance must be made for the drop across the air spaces. Carson has reported some tests on permeable substances in which from one-third to nine-tenths of the vapour pressure drop took place through the layer of air confined in the cell under the specimen. He also reported that in tests of materials very permeable to water vapour, including regenerated cellulose sheeting and textile fabrics, the vapour pressure drop through a half-inch layer of confined air was so great in comparison with that through the specimen alone that substantially the same value was obtained for each of the different materials tested. This problem is analogous to the heat transfer problem where allowance must be made for the surface-air conductance, but it can hardly be solved in the same way (by the use of a surface coefficient). It differs from the heat problem in that here the gradient is constant across the air space, while in the heat transfer problem the temperature is assumed constant at all distances from the surface greater than one-half inch.

Martley[10] in his investigations on the vapour permeability of wood calculated the resistances of the air spaces from the formula:

$$W = k \frac{u_1 - u_2}{l} at \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad [2]$$

where, W is the moisture loss in time t; l the distance between the surface of the liquid and the surface of the specimen; a the area; k the coefficient of moisture diffusivity in air; u_1 the moisture content of the air at the surface of the solution and u_2 that at the surface of the specimen. The values of k and u were obtained from Landolt-Börnstein Tabellen. In the case of the diffusion from the outer surface of the sample there is some difficulty in deciding the appropriate value to use for the area a, since it is not so clearly defined as in the inside of the cell.

Carson has suggested a method by means of which these pressure gradients in the air spaces may be eliminated. If the air can be made to move across the face of the sample with a high velocity it will be impossible for pressure gradients to build up. The minimum velocity which is necessary to eliminate all pressure differences has not been determined, but a fan blowing a strong stream of air across the face of the sample should be sufficient for the purpose. This can easily be arranged for the exterior surface of the specimen but is hardly available for the elimination of the gradient across the air space within the cell. The only solution which Carson has to offer in this case is to bring the surface of the liquid as close to the surface of the sample as possible. It is important that the liquid does not touch the surface as the permeability to liquid water differs from that for water vapour. To avoid this possibility Carson suggests that a solid desiccant be used in the cell and brought almost in contact with the surface of the test specimen. If these arrangements are used, no great error should arise in assuming that the vapour pressure drop across the specimen is equal to the overall pressure difference between desiccant and solution, but unless some such provision is made such an assumption is not justified. In any reliable measurements the vapour pressure drop through the specimen should be determined by subsidiary measurements. One method would be to use some substance which absorbs moisture such as wood or wood pulp. A determination of the amount of moisture absorbed at different positions through the sample together with the moisture absorption curve should be sufficient to determine accurately the vapour pressure gradient. Martley[11] has used this method in his determination of moisture migration in wood, and Taylor, Herrmann and Kemp[12] have employed the same procedure for rubber membranes. Abrams and Brabender[13] have used the absorption of filter paper to determine the relative humidity at intervals between the surface of the liquid and the test material. They have found that this varies very decidedly with the permeability of the material.

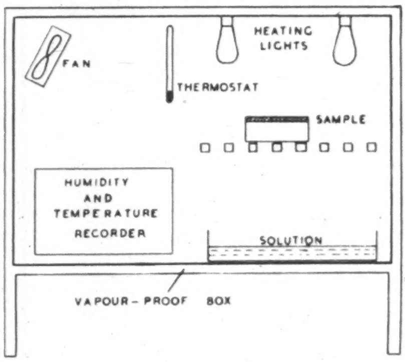

Fig. 3—Arrangement of apparatus for observing water vapour transmission

The arrangement of apparatus which is used in the National Research Laboratories is shown in Fig. 3. The box which is constructed of wood is painted on the inside with aluminum paint to prevent excessive loss of humidity. The temperature is kept constant by means of a thermostat controlling two lights. The humidity is supplied at a constant vapour pressure by a saturated solution in the tray on the bottom of the box. The relative humidity is recorded on a recording humidograph. A fan is placed so as to blow the air over the surface of the sample and to circulate air over the solution.

The items previously mentioned are the sources of error that are most apt to vitiate the measurements of the diffusion coefficients but there are several other factors which cannot be overlooked. It is, of course, essential to maintain a constant temperature as it is probable that the temperature has an effect not only on the vapour

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[10] Loc. Cit. See Note 8.

[11] Loc. Cit. See Note 8.

[12] Diffusion of Water Through Insulating Materials, by R. L. Taylor, D. B. Herrmann, and A. R. Kemp. (*Ind. Eng. Chem.*, Vol. 28, 1255, November, 1936.)

[13] Factors Affecting the Determination of Water Vapour Permeability, by Allen Abrams and G. J. Brabender. (*Paper Trade Journal*, 102 Tappi Sect. 205, 1936.)

Table 1—Values of Diffusion Coefficients for Various Materials

MATERIAL	THICKNESS TESTED (CM)	TEMPERATURE DEG C	VAPOUR PRESSURE DIFF. (MM HG)	d GRAMS/24 HOURS/SQ M/ $\frac{\text{MM HG}}{\text{CM}}$	D GRAMS/24 HOURS/SQ M/ MM HG	REFERENCE
Wood	From 1 to 5 cm	40	Average up to 20 mm	17.6[a]	....	Martley (8)
Asphalt Sealing Compound	0.0820	25	22.8 — 0.0	27.6×10^{-4}	0.034	Taylor, Herrmann & Kemp (12)
Wax	0.051	21.1	18.6 — 0.0	1.44×10^{-4}	0.0028	
Rubber	....	....	....	$(1.0 \text{ to } 240) \times 10^{-4}$	....	Carson (9)
Paper	....	....	....	....	73 to 250	Carson (9)
Waxed Paper	....	....	....	....	0.04 to 120	Carson (9)
Moisture-proofed Fiber board	....	....	....	....	0.2 to 2.0	Carson (9)
Protective Films (paints, varnishes and aircraft finishes)	....	....	....	....	0.19 to 26[b]	Carson (9)

[a]Mean value extracted from results given for Scots Pine Cylinder.
[b]These are for films 1 mil (0.0025 cm) thick.

pressure of the solution but also directly on the diffusion itself. There is still some doubt on this point; some experimenters[14] record an increase in water vapour permeability (per unit pressure difference) with increase in temperature, while others find no change except at high humidities and temperatures. However, it is probably quite legitimate to change data derived at one temperature to another standard temperature by correcting for vapour pressure alone as long as the temperatures are not widely separated.

Table of Diffusion Coefficients

In Table 1 some values of the diffusion coefficient are given. These values have been condensed from the table given by Carson, and his paper should be consulted for the original references. In this table, for some substances, only the range of values obtained is given since this will suffice to give an idea of the order of magnitude to be expected, and will, at the same time, indicate the variation in the different published results. A study of Carson's table indicates that there are almost as many different values of diffusion coefficient for a material as there are investigators resulting probably from both the variation in the material and the differences in the method of testing. It indicates the need for a systematic investigation leading to some standard method of test.

It is interesting to note that one published recommendation[15] for vapour seals indicates that the barrier used should transmit water vapour no faster than 0.40 g (grams) per square meter per hour under a vapour-pressure difference of 18 mm mercury.

Transforming these units to those referred to in Table 1, its value reduces to 0.53 g per square meter per 24 hours per millimeters mercury.

Conclusion

This brief outline of methods of measuring the diffusion coefficient has been presented in the hope that it may be of use in analyzing the problem of migration of water vapour in buildings. In the opinion of the author the measurement of the diffusion coefficients for various building materials is an immediate necessity in order to plan vapour-proof walls on sound principles. Some measurements are at present being made at the National Research Laboratories and it is to be hoped that other institutions will undertake similar investigations so that data from more than one source will be available.

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[14] Loc. Cit. See Note 12.

[15] Practical Aspects of Condensation in the Building Industry, by Engineering Division, Kimberly-Clark Corp.

NOMINEES FOR COMMITTEE ON RESEARCH

R. E. Daly
New York, N. Y.

W. A. Danielson
Louisville, Ky.

H. J. Rose
Pittsburgh, Pa.

C. Tasker
Toronto, Can.

C.-E. A. Winslow
New Haven, Conn.

Pittsburgh Plans for Big Annual Meeting of the Society

Westinghouse Memorial in Schenley Park

A FOUR-DAY meeting of the A.S.H.V.E. is to be held at the William Penn Hotel, Pittsburgh, January 23-26, and the Council has approved the plan of the Meetings Committee to place special emphasis on research activities.

This 45th Annual Meeting of the Society also marks the 20th year in which the Society has maintained its Research Laboratory at the U. S. Bureau of Mines. Many of the papers to be offered report the results of research investigations that have been under way this year. The main topics of discussion will be air flow, physiological sensations, heat transfer, fuels, and the design factors that influence air conditioning applications.

Some of the papers to be offered are:

Recent Advances in Physiological Knowledge and Their Bearing on Ventilating Practice, by C.-E. A. Winslow.
Cardiac Output, Peripheral Blood Flow, and Blood Volume Changes in Normal Individuals Subjected to Varying Environmental Temperatures, by F. K. Hick, R. W. Keeton, Nathaniel Glickman and H. C. Wall.
The Role of the Extremities in the Dissipation of Heat from the Body in Various Atmospheric and Physiological Conditions, by Charles Sheard, Marvin M. D. Williams, Grace M. Roth and Bayard T. Horton.
Skin Temperatures of the Extremities and Effective Temperature, by Charles Sheard, Marvin M. D. Williams and Bayard T. Horton.
Air Conditioning Requirements of an Operating Room, by F. C. Houghten and Dr. Lee Cook.
Some Reflection and Radiant Characteristics of Aluminum, by C. S. Taylor and J. D. Edwards.
A Study of the Heat Requirements of a Single-Glazed Test House and a Double-Glazed Test House, by M. L. Carr, R. A. Miller, Leighton Orr and David Shore.
Heat Transfer in Storage Water Heaters, by D. W. Nelson and A. A. Rosenberg.
Practical Methods of Preventing Condensation of Moisture in Walls, by F. B. Rowley, A. B. Algren and C. E. Lund.
Responsibility of A.S.H.V.E. Towards Solid Fuel Industry, by W. A. Danielson.
Selection of Solid Fuels from the Viewpoint of the Small Consumer, by P. Nicholls.
Small Stokers, by P. A. Mulcey and R. A. Sherman.
Improvements that are Needed in Heating Equipment, by T. H. Urdahl.
Performance of Stoker-Fired and Hand-Fired Warm-Air Furnaces in the Research Residence, by A. P. Kratz, S. Konzo and R. B. Engdahl.
Frictional Resistance to the Flow of Air in Ducts, by F. C. Houghten, J. B. Schmieler, John Zalovcik and Nicholas Ivanovic.
Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., and F. C. Houghten.
Limitations of Smoke Abatement Problems and Practical Methods for Their Elimination, by H. B. Meller.
Relation of Rank of Coal to Smoke Producing Tendencies, by H. J. Rose and F. P. Lasseter.
Air Filter Performance as Affected by Low Rate of Feed, Various Types of Carbon, Dust Particles Size and Density, by F. B. Rowley and R. C. Jordan.
The Effect of Heat Storage and the Variation in Outdoor Temperature and Solar Intensity on the Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan and F. O. Urban.
The Determination and Practical Application of Weather Design Data, by T. H. Urdahl, John Everetts, Jr., and J. C. Albright.
Influence of Building Construction on Air Conditioning, by F. E. Giesecke.

The Committee on Arrangements of the Pittsburgh Chapter, headed by R. A. Miller, is planning a fine program of entertainment for members and ladies who are present. All members will find it worthwhile to attend the 45th Annual Meeting, January 23-26, 1939. Pittsburgh says Welcome A.S.H.V.E.

Pittsburgh Chapter Plans for A.S.H.V.E. Annual Meeting

October 10, 1938. Pres. J. F. Collins, Jr. called the Pittsburgh Chapter meeting to order at 7:35 p.m. in the private dining room of Stouffer's Restaurant. An attendance of 48 members and guests was recorded.

The minutes of the previous meeting were read and approved and reports were then in order.

Chairman P. A. Edwards of the Membership Committee introduced H. J. Kirkendall as a new member.

Due to the absence of R. A. Miller, general chairman of the Committee on Arrangements for the Society's 45th Annual Meeting in January, 1939, President Collins presented his report, which included a tentative budget of income and expenditures previously approved by the Board of Governors.

President Collins appointed the following members to the Chapter's Nominating Committee: M. L. Carr, *chairman;* R. J. J. Tennant.

R. H. Heilman, senior industrial fellow, Mellon Institute, was introduced by President Collins as the speaker of the evening. Mr. Heilman gave a resumé of his research in the field of heat transfer through pipe and building insulating material. His paper, which was well illustrated by slides, pointed out the effect of density at low and high mean temperature on heat transfer, the relative importance of radiation and convection at different temperature levels, and the effect of shape on rate of heat transfer. Improvements in the design of insulating materials which resulted from this research were also described. Mr. Heilman concluded his lecture with a description of new ducts fabricated from asbestos paper.

According to Secy. T. F. Rockwell's report, the paper proved to be very interesting to the members and guests, and the meeting was adjourned at 9:50 p.m.

September 12, 1938. Pittsburgh Chapter opened the season of 1938-39 with a meeting in Stouffer's Restaurant, which was called to order by Pres. J. F. Collins, Jr. Fifty-two members and guests were present.

L. S. Maehling, treasurer, presented a report giving the Chapter's financial condition as of September 1, 1938.

President Collins introduced R. D. Van Houten as the speaker. Mr. Van Houten is vice-president of the Sales Analysis Institute and chose as his subject Engineering Sales Presentation. In his introduction the speaker stressed the point that selling is the income side of business and that everyone must sell at some time. Mr. Van Houten then analyzed in detail the necessary steps in a well planned sales campaign.

The meeting was adjourned at 10:00 p.m.

F. B. Rowley Speaks at Michigan Meeting

October 10, 1938. The first meeting of the 1938-39 season of the Michigan Chapter was held at Huyler's L'Aiglon in the Fisher Building with approximately 80 members and guests present.

Pres. F. J. Linsenmeyer called the meeting to order at 7:30 p. m. and asked for roll call, reports of the secretary and treasurer, and the chairmen of the Membership and Program Committees.

Tentative plans for the Semi-Annual Meeting of the A. S. H. V. E. to be held at Mackinac Island, July 2-5, 1939, were outlined by W. G. Boales.

Prof. F. B. Rowley, past president of the Society and director of the Engineering Experiment Station, University of Minnesota, was the guest speaker of the evening and visited the Chapter through the Society's Speakers Bureau.

In an illustrated lecture Professor Rowley presented information resulting from research conducted on the problems of condensation within walls. He pointed out that condensation within walls was not necessarily a function of the type or make of insulation used, but was more directly dependent upon the vapor seal used in the wall. Professor Rowley also showed that condensation in attic spaces could be eliminated by proper attic ventilation.

After the lecture, Edmund Kuhlman, president of the *Greater Detroit Home Builders Association*, presented the problem of condensation and insulation from the builder's point of view. Prof. L. G. Miller, head of the Department of Mechanical Engineering, Michigan State College, discussed some research being conducted on vapor transfer through various typical wall sections. Joseph Wolff, Commissioner of Building and Safety Engineering, City of Detroit, told of the interest of his department in condensation and its effect upon wood frame decay.

The meeting was adjourned at 10:30 p.m. with a rising vote of thanks to Professor Rowley, according to the report of Secy. G. H. Tuttle.

Gas-Fired Refrigerating Units Interest Southern California Chapter

October 12, 1938. The first meeting of the 1938-39 season of Southern California Chapter was held at the Colonial Banquet Room, Los Angeles, with an attendance of 66 members and guests.

After the meeting was called to order, the new officers were introduced and the following committee chairmen were appointed: *Membership Committee*—O. W. Ott; *Publicity Committee*—H. H. Douglas; *Speakers Committee*—E. H. Kendall; and *Program Committee*—J. F. Park.

The annual report was read by the secretary and the treasurer's report was given.

Glen Miller of Southern Counties Gas Co., was introduced as the speaker of the evening and discussed Small Gas-Fired Refrigerating Units for Air Conditioning. Mr. Miller's talk was very interesting and well presented, according to Secy. A. J. Hess, and included a brief resumé of general types, followed by a detailed description of the more important systems on the market with their advantages and disadvantages. The economics of the use of gas for cooling as compared to the use of electricity was discussed, and comments were offered on possible future developments.

After the conclusion of Mr. Miller's talk, an interesting group discussion was conducted before the meeting adjourned.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 26 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by November 15, 1938, these candidates will be ballotted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
CRONEY, P. A., Chief of Mech. Section, U. S. Housing Authority, Washington, D. C.	A. F. H. Scott T. H. Urdahl	C. E. Mayette V. F. Manos (*Non-Member*)
CULLIN, W. W., Chief Engr., Home Insulation Div., Johns-Manville Sales Corp., New York, N. Y.	L. G. Miller F. B. Rowley	Elliott Harrington Wharton Clay